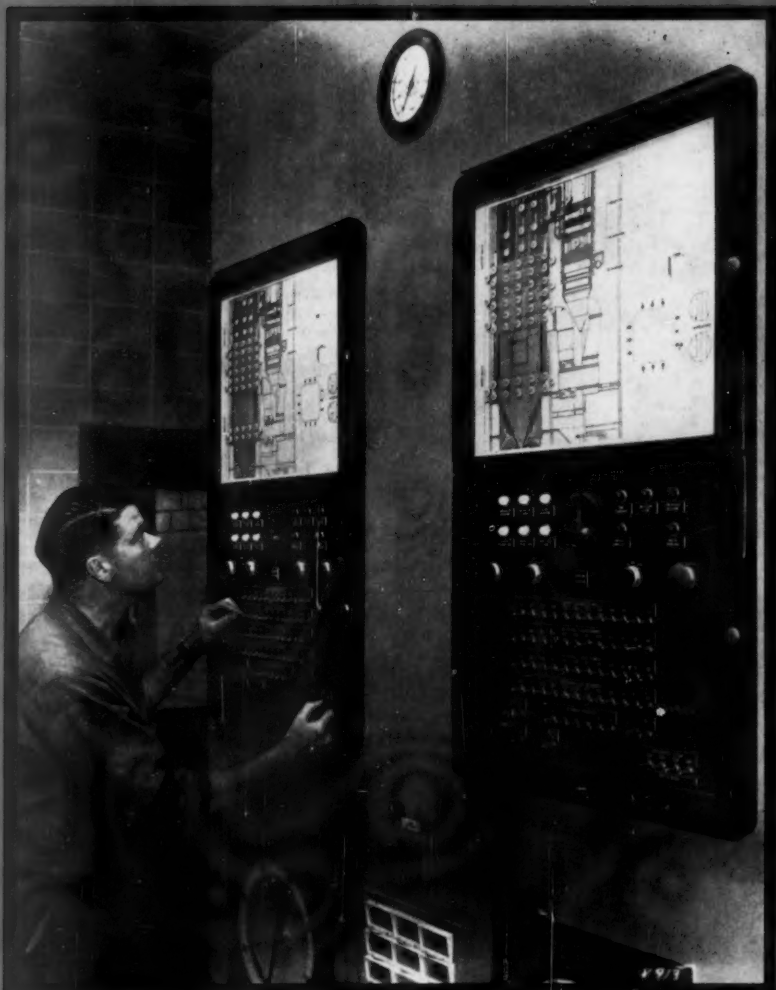
COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

January 1960



Soot blowing operations for the Eddystone Station of the Philadelphia Electric Co. is centered in these panels. For more about Eddystone see pp. 32-36.

Flow of Coal in Hoppers

Space Age Hydrostatic Test

Hyperbolic Cooling Towers

COMBUSTION

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Feature Articles

Editorials

Departments

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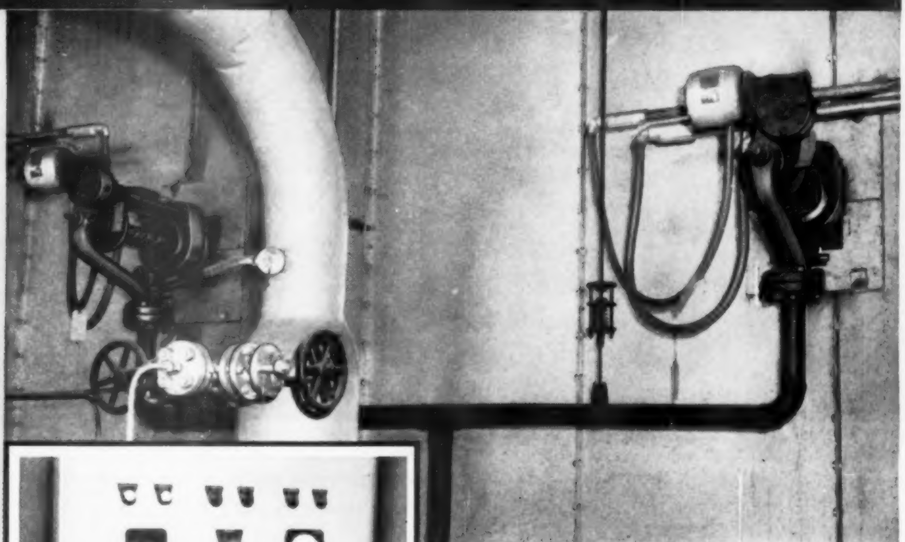

Two of the air blowing Model IR short retracting wall blowers with electric automatic operation.

Diamond Automatic Sequential Control Panel installed as part of the modernization program at the F. M. Tait Station. This panel automatically operates each blower in turn until the blowing cycle is complete. Any individual blower may be operated independently by simple turning switches. Cleaning program is set up by switches. Operator starts — thenceforth operation is automatic.

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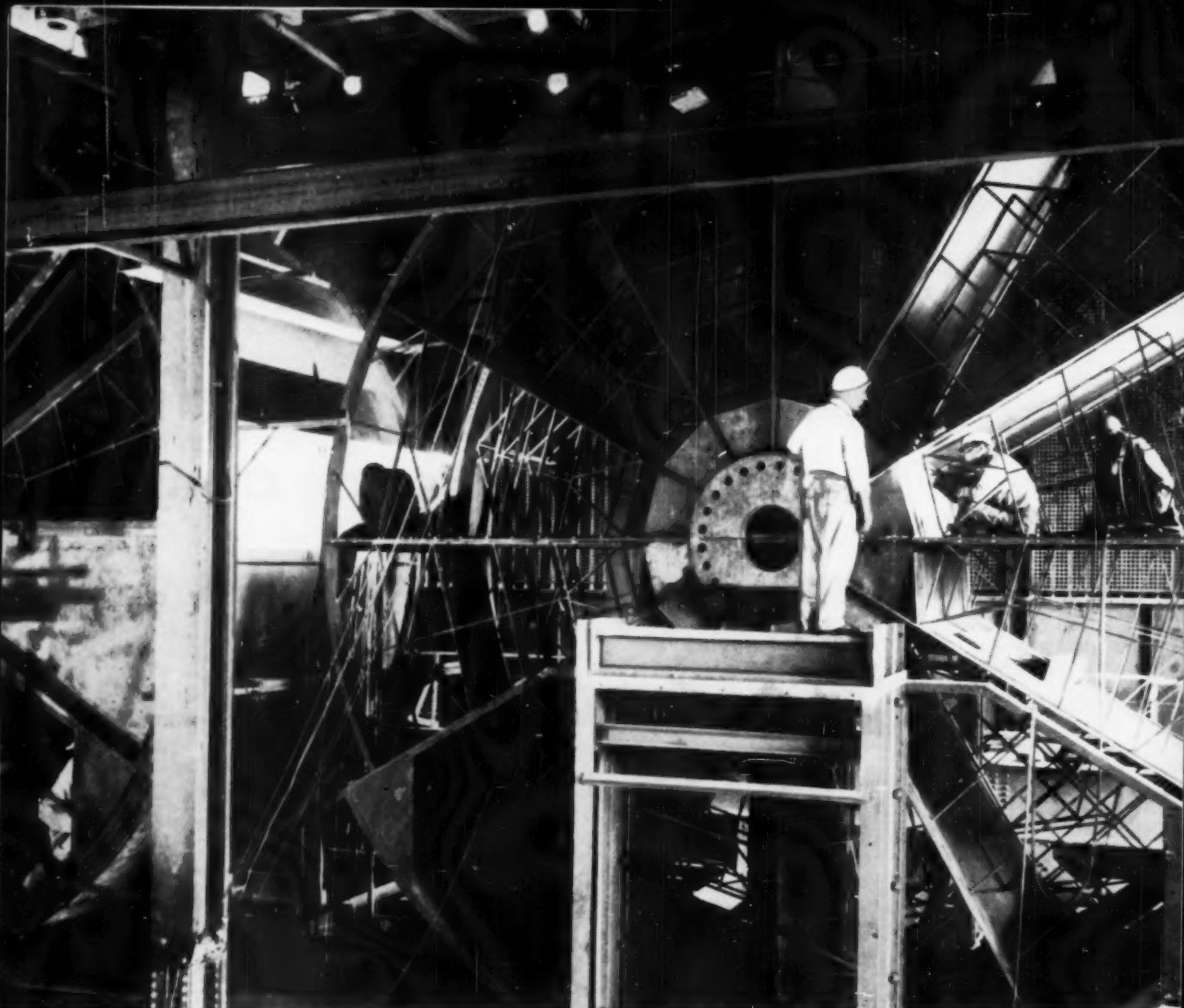
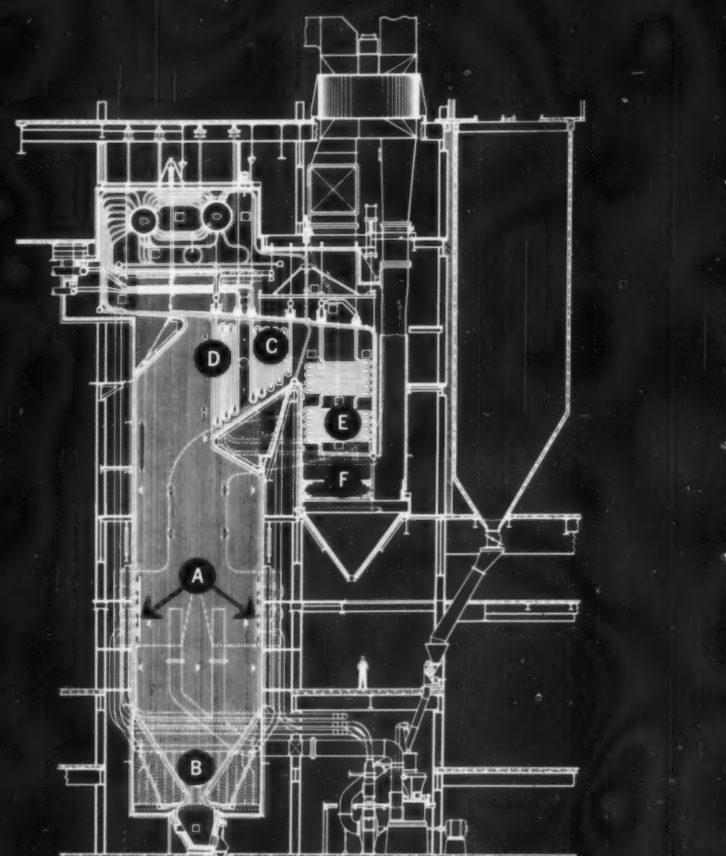


Two of the four Ljungstrom Air Preheaters under construction at Potomac Electric Power Co.'s Dickerson Power Plant, Dickerson, Md. Each pair of these Ljungstroms, with a combined heat exchange surface of 475,000 sq. ft., will serve one boiler, evaporating 1,300,000 lbs of steam/hr. The first 175,000 KW unit was placed in operation in June this year and a duplicate unit is now under construction.

A 10-year progress report on...

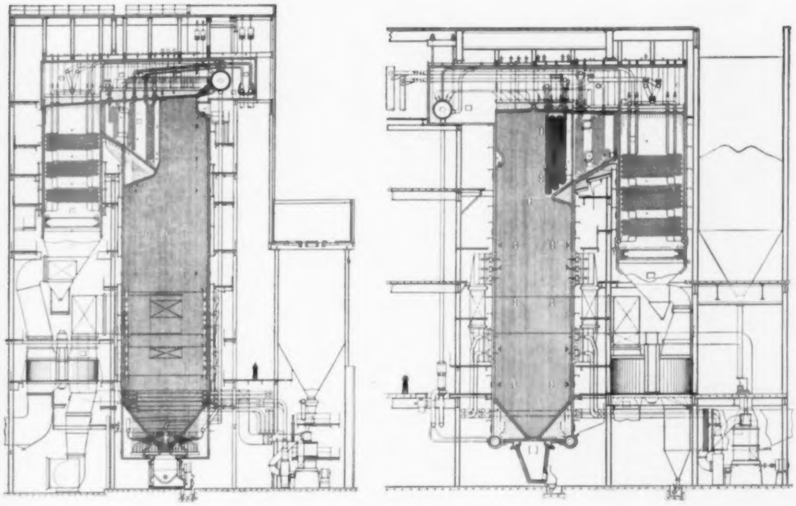
What Reheat has contributed



This unit, placed in service at the Edgar Station of the Boston Edison Company in 1949, represented the original C-E design concept of a properly arranged reheat boiler. Its chief characteristics were: tangential firing A with tilting burner nozzles to control reheat and superheat temperatures; a dry bottom furnace B; reheater surface C between the finishing superheater D, located at the furnace exit, and the primary superheater E in the back pass; economizer surface F below the primary superheater. For the period from August, 1949, to December 31, 1958, this unit had an average availability* of 94.5% and an average capacity factor** of 98.4%.

Average Availability—In service or available. Not considered available while down for inspection or repair, or while in the process of starting up or shutting down.

***Average Capacity Factor—Ratio of average hourly output—net kw—to rating of turbine-generator.*



These two recent reheat designs (left — natural circulation; right—controlled circulation) demonstrate the basic similarities between Combustion's first post-war reheat design, as represented by the Edgar installation, and its present-day reheat designs.

to Reduced Power Costs

The average rate of fuel consumption per kw-hr over the past decade has decreased from 1.30 lb in 1948 to .905 lb in 1958–a reduction of about 30 per cent. While an important part of this economic gain must be credited to the adoption of higher steam pressures and temperatures, the principal part has resulted from the widespread adoption of the reheat cycle.

C-E's role in the development and application of post-war reheat boiler design has been a major one. The first C-E unit of this design, ordered in 1947 by Boston Edison Company† for its Edgar Station, went into service in August, 1949. As of now–a decade later–a total of 279 reheat units, including 119 of the controlled circulation design, have been ordered by American utilities for an aggregate capacity of 39,100,000 kw. This is equivalent to more than one-third of the total steam-generated capacity of the utility industry as of the first of this year. Of the 279 units, 208, with a capacity of over 26,000,000 kw, are in service.

System generating costs are importantly affected by continuity of service of the more efficient units. The remarkably fine performance of C-E Reheat Units in this respect is evident from the records of 150 units, on which data are available from start-up dates to the end of 1958. The composite record of these units shows an average availability of 95.07% and an average capacity factor of 92.9%. The operating records of these units for the periods covered add up to a total of 560 boiler-years of service.

The consistently good performance records of C-E reheat installations is primarily attributable to the soundness of the original design concept. While numerous refinements and improvements of design detail have been made through the years, the basic arrangement of principal components has remained the same as evidenced by the accompanying drawings of the first unit (Edgar Station) and two recent units.

†Boston Edison also pioneered in the early development of the reheat cycle, having installed the country's first 1200-psi reheat boiler in 1925.

COMBUSTION ENGINEERING

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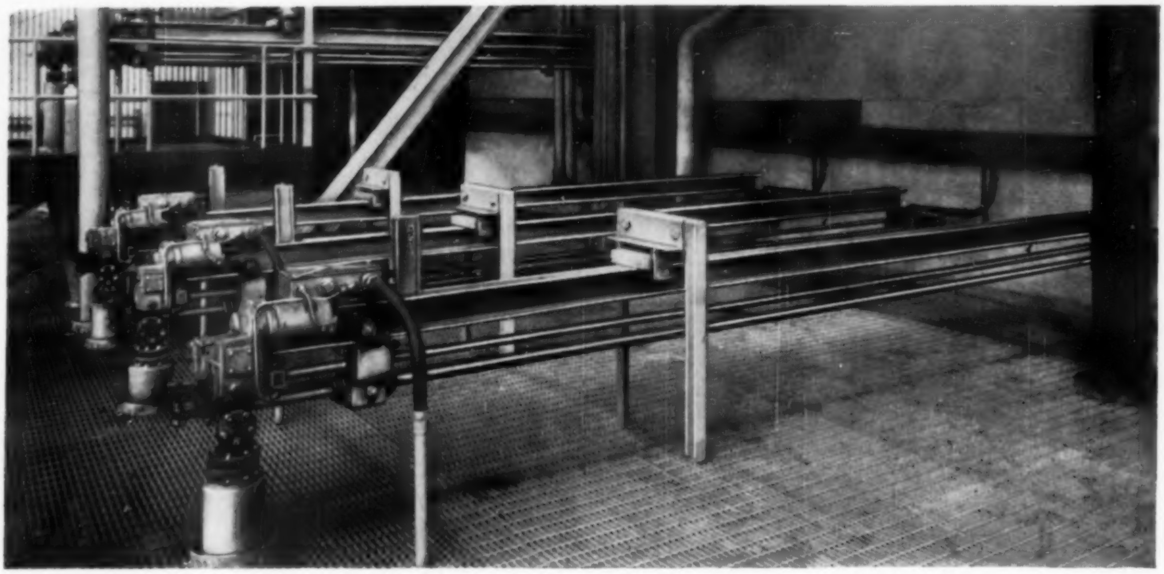
C-249

ALL TYPES OF STEAM GENERATING, FUEL BURNING AND RELATED EQUIPMENT; NUCLEAR REACTORS; PAPER MILL EQUIPMENT; PULVERIZERS; FLASH DRYING SYSTEMS; PRESSURE VESSELS; SOIL PIPE



Vulcan Automatic-Sequential System controls soot blowing at new Gannon Station

As the first modern utility boiler in Florida to be fired by coal, Tampa Electric Company's Gannon Station incorporates the most advanced control systems. Unit 1, with its boiler rating at 950,000 pounds per hour at 1760 psig and 1000 degrees F., uses a Vulcan soot blowing system that features both group and unit control.

Vulcan T-3-E long retractable soot blowers with their dual-motor drives clean Gannon's Unit 1 boiler with double-helix patterns for complete coverage. The system includes Vulcan wall deslaggers and rotary soot blowers—all driven by electric motors and using steam as the blowing medium.

A Copes-Vulcan diaphragm valve reduces steam pressure for blowing and a Copes-Vulcan motor-operated valve is used for shut-off.

A similar Vulcan system will be installed on Gannon's Unit 2, rated at 950,000 pounds per hour at 1760 psig and 1000 degrees F.

Besides fully-automatic soot blowing systems, Copes-Vulcan offers complete systems for controlling combustion, feedwater, boiler feed pump recirculation, and steam temperature. Whether your boiler is large or small, power or process, Copes-Vulcan can provide a unit or an integrated package, custom designed to your requirements.

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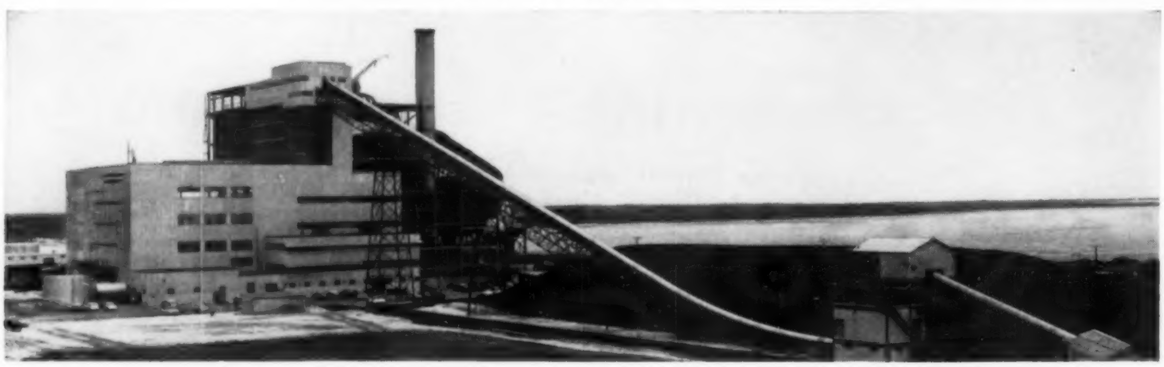
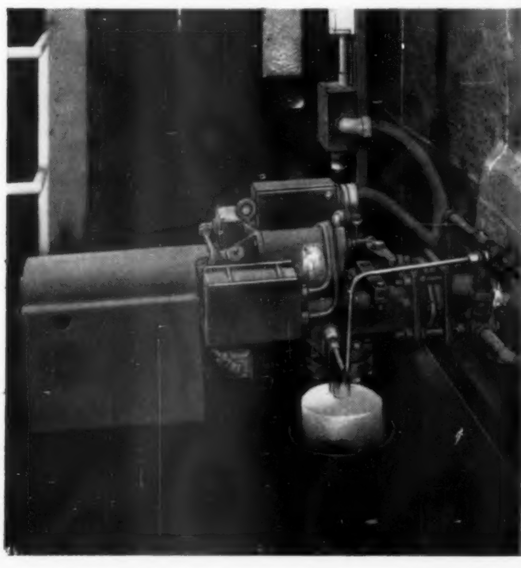
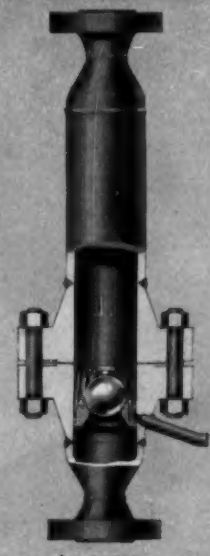


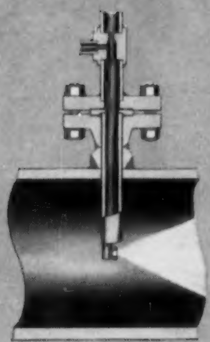
Finger tip control for Gannon's operators . . . with Vulcan's Automatic-Sequential group and unit control. Operator can preselect any number of soot blowers for automatic sequence . . . preselect any number of units or groups of units to be operated individually . . . or switch instantly to single unit operation. Program lights and toggle switches are located on engraved boiler diagram. Automatic-Sequential systems use steam, air or a combination as the blowing medium without a change in equipment. Write for Bulletin 1029.

Wall deslagger conserves steam . . . minimizes average slag thickness. Vulcan's RW-3E is equipped with dual motors: one to traverse *fast* to get the nozzle to and from the blowing position almost instantly . . . the other to rotate the tube slowly for thorough cleaning. The RW-3E may be installed indoors or out, at almost any angle from side wall, roof or floor. It has only one outside stuffing box, no threads or working parts in the blowing medium to assure easy maintenance. Write for Bulletin 1034.



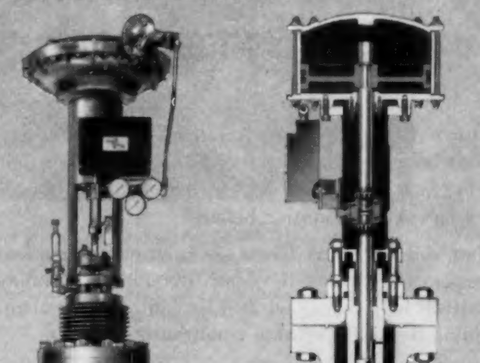
C-V NEWS NOTES





Desuperheaters improve temperature control. Variable-Orifice Desuperheater* (left) uses a weighted steel ball to control orifice opening . . . holds reduced steam temperature constant only twenty feet downstream from desuperheater outlet, even over a 50-to-1 load range. Bulletin 1037. *Patent Applied For

Steam-Assist (right) meets all specifications for conventional steam-atomizing type yet normally uses assisting steam only on light loads where control is most difficult. Mechanical-atomizing types also available. Bulletin 1024-A.



Versatile regulator valves offer new simplicity of design. Diaphragm-type CV-D (left) is designed for remote control service. It can be direct or reverse acting, has excellent rangeability. Piston-type CV-P (right) is designed for high-duty service . . . assures maximum power with precise positioning.

All Copes-Vulcan valves are tailored to the job. The style of valve port is selected to provide optimum control for specified operating conditions. Bulletin 1027.

8686 Millivolt Potentiometer

Make speedier checks of recorders, controllers and base or noble metal thermocouples in industrial plants with the new three-dial **8686 Portable Millivolt Potentiometer.** Features such as a central reading window . . . where measured values appear as a row of digits with a scale interpolation . . . simplify calibration of thermocouples and test measurements. The 8686 Potentiometer has: a wide operating range of -10.0 to $+100.1$ mv and $+1010$ to $+1020$ mv for standard cell calibration; and a high accuracy of $\pm$(0.05% of reading $+3\mu v$) without reference junction compensation, $\pm$(0.05% of reading $+ 6\ \mu v$) with ref. jct. comp. Write for Data Sheet E-33(1A).

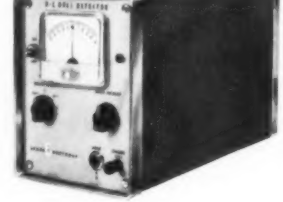
8690 Millivolt Potentiometer

Fast, accurate in-plant checking of thermocouples, recorders and controllers is expedited by the new 8690 Portable Millivolt Potentiometer. Design features such as "In-line Readout", which permit measured values to be read in digits with a scale interpolation, accelerate temperature measurements. The 8690 provides a wide range of -11.0 to $+101.0$ mv which eliminates input reversals; an accuracy of $\pm$(0.05% of reading $+ 20\ \mu v$) without reference junction compensation, $\pm$(0.05% of reading $+ 40\ \mu v$) with ref. jct. comp. Write for Data Sheet E-33(1B).

8692 and 8693 Temperature Potentiometers

If you want to make a variety of temperature measurments quickly with one flexible instrument, investigate the new time-saving 8692 Single-Range or 8693 Double-Range Temperature Potentiometers. Available in any of 24 interchangeable temperature and millivolt ranges, these instruments read directly in degrees F or C on a scale 27½" long. Convenience features include: simplified range changes . . . only a screwdriver is needed to change a circuit panel, scale and binding post studs; automatic reference junction compensation . . . reference coil, built into circuit panel, compensates for thermocouple being used; accuracy . . . $\pm$0.2% of range. Write for Data Sheet ND42-33(1A).

Need a fast-operating, high-sensitivity, high-quality null indicator for use in research, testing and production checking? Here's a new 9834 Guarded D-C Null Detector having a short period of less than two seconds for source resistances up to 1000 ohms, increasing to 4 seconds at 100,000 ohms . . . ideal for measurements with guarded or unguarded potentiometers and bridges. Of rugged construction, this portable, line-operated detector provides numerous convenience features which include four degrees of sensitivity, with a basic sensitivity of 0.2 μv/mm (0.3 μv/scale div.), and a noise level of less than $\pm 0.1\ \mu v$. Write for Data Sheet ED7(2).



9834 Guarded D-C Null Detector

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Hall Industrial Water Report

VOLUME 8 JANUARY 1960 NUMBER 1

Don't Carry Things Too Far

This familiar expression predicts trouble. Just as trouble with deposits comes when steam carries droplets of boiler water too far—out of the boiler steam drum into superheaters, turbines, heat exchangers and heating systems. Superheater tubes burn out, turbines lose efficiency and capacity, and heat transfer suffers.

Hall engineers know how to combat carryover problems. They are familiar with the mechanical methods of purifying steam and know how to determine whether antifoams are also needed. They can help you to improve steam quality just as they can serve you on other water problems in the steam plant, or in cooling process or waste water disposal facilities.

Spreading Oil on the Water

At a southern synthetic fiber mill serious fouling of a turbine with water-soluble deposits required washing at intervals of approximately three months. This puzzled Hall engineer T. A. McConomy no end because normally his checks on steam showed it to be of good quality. Furthermore, he could find no place in the mill where the condensate, makeup water or boiler feedwater was being contaminated. The use of antifoams in the boiler water did not improve conditions.

One day heavy carryover occurred—so heavy that steam temperature temporarily dropped almost 100°F. The alert operators collected samples of boiler water. Analysis showed one to contain 28 ppm of ether-extractable material and the other an astounding 1185 ppm. Infrared analysis showed the ether-soluble material to be largely lubricating oil.

What was the source of the oil? Previous investigation had ruled out the mill. McConomy asked the operators to keep a careful watch on the river water at the plant intake. A few days later an oil slick showed up.

It was now easy to trace the source of contamination upstream to several service stations and small manufacturing plants which were dumping oil into the river. The job of getting them to stop was harder. So the filter plant operator now has an extra duty called "river watching." This makes possible the diversion of oil-contaminated water from the steam plant, and saves the cost of additional, expensive equipment.

Baffling Leaks

Carryover was so bad in a midwestern automobile body plant that steam traps were continually plugging and a turbine-driven air compressor couldn't produce at rated capacity. It was determined that one boiler of three was causing all of the trouble.

Each boiler was put through its paces while steam quality was checked conductometrically by Hall engineer L. P. Dougherty. Steam from two of the boilers was found to be excellent under all conditions, even when operated at ratings well above normal. The third boiler acted all right until the rating was pushed a little above normal, then gobs of water went out in the steam.

Having proved which boiler was the bad actor, Dougherty asked that it be taken out of service for inspection of steam drum internals. Sure enough, the baffles were found to be hanging loosely, leaving spaces where steam could short-circuit to the offtakes without following the designed path.

Baffles were tightened and the leaks closed up. Additional steam purity checks showed that the repairs were effective. Trouble stopped and has not recurred.

Embrittlement in Steam Line

Failure of steam lines due to caustic embrittlement is almost unheard of. It can occur, however, if the necessary conditions are present. Caustic soda must get into superheated steam to produce a concentrated solution and this solution must contact highly stressed metal.

In one case high-pressure steam was reduced to 60 psi and desuperheated to within 40°F of saturation temperature. Desuperheating water was a mixture of condensate and deionized water. The pH value of the deionized water was adjusted to 9.0-10.0 with caustic soda for protecting uncoated steel pipe from corrosion and for establishing desired alkalinity in the boiler water.

The steam line was welded. Chill rings had been used and there were open spaces between the rings and the internal surface of the line. When desuperheating water was trapped in the spaces, it concentrated to produce a strong caustic solution in contact with the stressed metal in the welds. Cracking and failure occurred.

In this instance the major source of caustic soda was that added to the deionized water. However, sodium leakage or insufficient rinsing after anion exchanger regeneration could supply the caustic soda. Obviously then, deionizers must be operated with this in mind if the water is to be used for desuperheating.

Water is your industry's most important raw material. Use it wisely.

Industrial Water Problems Require Special Handling

There are no "stock answers" to industrial water problems. For information on how the Hall System can help you solve your particular water problems, write, wire or call address below.

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COMBUSTION

Editorial

Think

There is no more gratifying or perfect a process involving the individual alone, than the neglected art of thinking. Engineers who practice this process as the means of earning their daily bread derive immense satisfaction from it. Indeed it is the reasoning process rather than any external symbols or kudos that mark the engineer as a truly professional man. Consider the medical man—when he assembles and verifies the patient's symptoms and conducts his preliminary tests he is gathering and assigning relative importance to a body of facts. When he reasons from these facts to a conclusion (diagnosis), he has performed the most richly rewarding act his profession offers. The subsequent operation, no matter how much drama is imputed to it by TV and moving pictures, is but the act of a skilled technician.

It seems to us that professional status cannot generally be accorded to engineers until they renounce the technician's label and commence reasoning on a plane expected of professional people. Those that will question the need for this change need only examine some current technical articles and papers. How often we observe an author lead us through a series of assumptions only to proceed, "Now from these *facts* it becomes obvious, etc." How tragic that the reasoning process has broken down before it had even begun. The confusion in the use of words and the preoccupation with technical jargon noted in most technical articles betrays a confused state of mind wholly inconsistent with clear and logical thought.

While it is not our purpose to add another voice to the outcry against engineering and scientific education in this country—this problem in fact appears to have an international scope—we do think it important to state that all education should have as its *primary* goal the inculcation, development and polishing of the reasoning process. Failing this we can teach only by rote and thereby develop in the main the type of scholar Montaigne described when he wrote, "See but one of these our university men or bookish scholars return from a school after he hath there spent ten or twelve years under a pedant's charge. Who so inapt for any matter? Who so unfit for any company? All the advantage you discover in him is that he is more sottish, more stupid and more presumptuous than before he went from home."

It is not to be expected that there will be any outcry that crowded curricula do not allow the student to learn to think—so monstrous a perversion of propriety would be unthinkable. The great number of engineers who have succeeded in fields other than their university specialty attest to the facility with which the techniques and practices of a particular branch of engineering may be acquired. As we approach the birthday of Abraham Lincoln it is perhaps appropriate to note that with no formal training but with an innate ability to reason he taught himself the mathematics and techniques of surveying in six weeks, studying only at night. Nor is it inappropriate to observe that in his only recorded "mistake" he laid out a crooked street so that an impoverished widow and her children would not be displaced from their home—a touch of humanity in a logical, mathematical process.

Over a decade ago, Prof. K. E. Smith of the University of Detroit astonished a class of senior engineering students by announcing that 50 per cent of their final examination grade would be determined by the manner in which they reasoned the approach to a highly technical problem, the solution of which was beyond their capability. Certainly there must be many other engineering educators who feel the fundamental importance of thinking in engineering education, but evidence of an inexorable movement toward this end has so far escaped us. It is our sincere belief that such a trend would provide benefits of inestimable value to the engineering student and to the engineering profession.

This article presents (1) theories on the flow of bulk materials in Hoppers and (2) the Hyperbolic Hopper Outlet derived from those theories in order to eliminate clogging of coal discharge from the hopper.

By Yee Lee†
Bin-Dicator Co.

Flow of Coal in Hoppers*

CLOGGING of bulk materials in hoppers is a costly problem in many industries. The presence of this trouble wastes large amounts of manpower, disturbs the operation of an automatic feeding system and reduces production. It is especially true in the coal storage system of steam power plants.

The trouble in Peipu Power Station of the Taiwan Power Company, Taiwan, China, where the author was station supt., is one of the more serious examples. The bunkers for the two boilers built in 1939 have a capacity of 250 cubic meters, with bottom sloped at 55 deg. The coal stored in these bunkers would not discharge itself as expected. In order to overcome this difficulty two men were assigned to the hopper outlet level to poke or pound and two more on the top of the bunkers to break the ratholes. With such an arrangement, it was still impossible to get enough coal into the pulverizers if the coal were really wet. As soon as the coal ceased to flow, power output, of course, declined immediately. Sometimes the fire consequently went out in the furnace. At this moment the operators, besides restoring coal flow, had to reignite the fire and hurry to adjust the combustion and load. We have several cases in which the power output was cut down to one third of the plant capacity.

In 1953, two more boilers were installed in the same plant. Owing to the previous experience, special considerations were taken in the design of coal bunkers. The hoppers were composed of 70-deg inclination walls and the outlets were lined with stainless steel sheet. Besides, two electric vibrators were attached to each hopper, one at the outlet and one on the chute. This construction it was felt represented the best engineering knowledge available at that time. However interruption of coal flow still existed from time to time. Hammering and poking were still frequently needed.

The problem was referred to consultant engineers and manufacturers. Several corrective suggestions were tried but none of them proved satisfactory. Finally we took up the problem ourselves. Eventually some principles concerning coal flow in hoppers were found and while investigations are incomplete a practical solution has been achieved. Although the findings are not complete and some data not evaluated, a remedial device designed on these principles did solve our problem.

* Patents on Mr. Lee's invention are pending.
† Engineering Dept.; On leave from Taiwan Power Co., Formosa.

Coal Flow Characteristics

For a certain height of coal column or other bulk materials, the pressure against the wall varies with the slope of wall. Generally speaking, the pressure is maximum in a vertical downward direction, and decreases when it is deviated from the vertical.

Letting P denote the pressure in a vertical downward direction and P' the pressure at an angle of θ from the vertical, the ratio

$$\eta = \frac{P'}{P} \tag{1}$$

is called the inclination factor for angle θ, Fig. 1-1.

The coal pressure in different directions is measured with the device shown in Fig. 1-2. The pressure-indicating element consists of a capsule with a rubber diaphragm as the sensing element and a copper tube connecting the chamber of the glass tube which is standing in a bottle of colored water. The support base of this element is bolted to the wall of a coal container and the element can be rotated around a horizontal axis through the surface of the capsule diaphragm.

First, with the diaphragm facing upward, coal is put into the container gradually till it reaches a certain

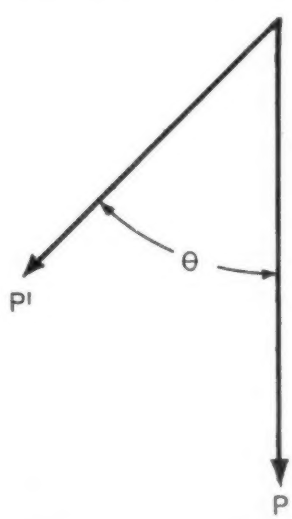


Fig. 1-1 Coal pressure at an inclination from the vertical

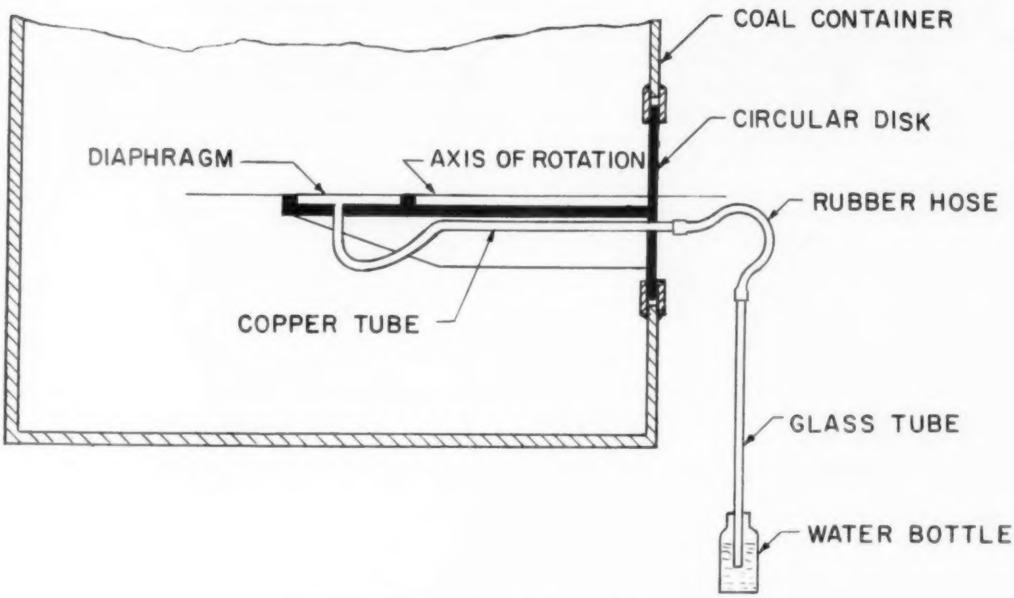


Fig. 1-2 Apparatus for measuring coal pressure in various directions

level. The diaphragm now deforms under the influence of coal pressure against it. Deformation of the diaphragm pushes air through the copper tube to the glass tube which indicates the pressure by the change in level of the colored water column. Then the pressure-indicating element is rotated 10 deg and the measurement is repeated. The measurement of loose coal pressure in different directions together with the calculated inclination factor are plotted as shown in Fig. 1-3.

Coefficient of Friction

The frictional resistance of cohesionless coal particles moving on a smooth wall depends on the contacting pressure normal to the wall and the coefficient of friction between wall and particles. The coefficient of friction is equal to the tangent of the angle of slide of the material with the wall surface. Let f_1 be the coefficient, α the angle of slide, Pn the normal pressure and R_1 the wall friction resistance, then $f_1 = \tan \alpha$, $R_1 = Pnf_1$.

Similarly the frictional resistance of cohesionless coal particles moving against a surface of similar coal particles depends on contacting pressure and the coefficient of friction which is equal to the angle of internal friction or angle of repose for the coal. Let R_2 by the resistance of internal friction, f_2 the coefficient and ϕ the angle of repose, then

$$f_2 = \tan \phi \qquad (2)$$

$$R_2 = Pnf_2 \qquad (3)$$

Coal particles are cohesionless when the surface is dry. If the coal gets wet, it becomes cohesive and an additional resistance besides the frictions would appear.

Compaction

Coal is usually compacted due to one or more of the following causes:

1. Pressure due to the weight of overlying materials.
2. Impact of falling materials feeding to the hopper.
3. Vibration of the vessel.

Coal in the loose state has voids or pores between particles which may occupy 40% of the total volume. When the coal is compacted, the total area is reduced due to closer arrangement of particles. Then the movement of individual particles is more or less restricted by adjacent particles and the internal friction of the material increases even after the causes of compaction are removed. Some wet coals under high compression may be solidified into a rigid body. Therefore compacted coal cannot flow as freely as coal in the loose state. Once the coal body deforms in the course of flow however, compactness due to impact and vibration relieves itself.

Coal Flow Resistance in Hoppers

For dry coal flowing down through a vertical cylindrical vessel which has uniform cross-sectional area, the only resistance to flow is the friction between the wall and the coal in contact with it. The wall friction per unit area of contact is proportional to the coefficient of friction and the normal pressure of the material against the wall. The magnitude of wall friction alone, however, does not give a true picture of flow characteristics of coal in the vessel. The best criteria to indicate the ability of the coal to flow in this case is the resistance force per unit volume of coal, or:

$$Z_1 = \frac{f_1 P_\eta U \Delta y}{A \Delta y} = f_1 P_\eta \frac{U}{A} \qquad (4)$$

Where:

f_1 = coefficient of friction
P = pressure at the elevation

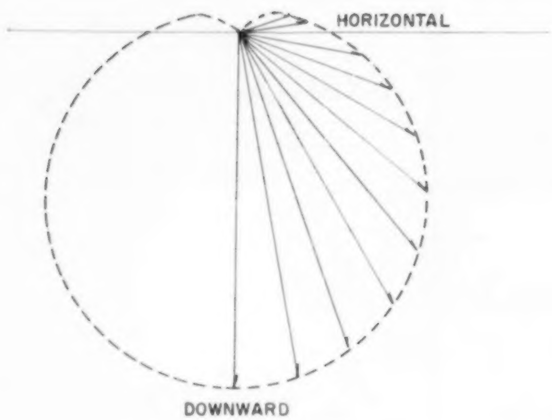


Fig. 1-3 Distribution of coal pressure

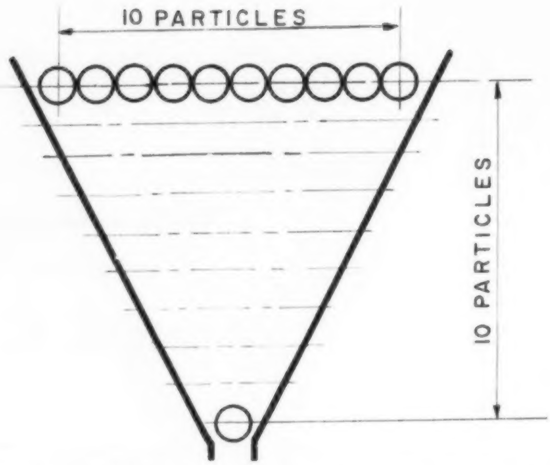


Fig. 2-1 Hypothetical hopper showing arrangement of coal particles

U = circumference of the vessel
A = cross-sectional area
η = inclination factor
Δy = small vertical distance
Z_1 = resistance force per unit volume of material
W = density of the material

If the resistance Z_1 is less than the gravity force, the material will flow down, and the downward pressure will be

$$P = Wy - \int_0^y P_\eta f_1 \,(U/A)\, dy \qquad (5)$$

For a vertical cylindrical vessel and radius (r), equation (5) leads to the solution:

$$P = \frac{Wr}{2f_1\eta}\,(1 - e^{-2f_1\eta y/r}) \qquad (6)$$

This is the Jannessen equation. Actually pressure on the same elevation is not uniform but higher at the center and lower near the wall. However the result from this equation is fairly close to the average pressure on the cross-section area.

For the flow of dry coal in hoppers, the resistance is different from that of cylindrical vessels. For example, certain coal flow rates could easily be accommodated by 6-in. pipe but cannot be handled by a cone-shaped hopper of the same size discharge opening. The conical hopper has a wider average passage area and smaller wall friction factor $\eta(U/A)$. It should have less wall frictional resistance per unit volume of coal. Why then should its flow capacity be less? It is believed that for the flow of coal in a hopper, there must be another component of resistance besides the wall friction.

Let us consider the flow of solids in a hopper of square section as shown in Fig. 2-1. On the top layer there are 10 particles on a row and 100 particles on a layer. When the material flows down to the middle section, each layer can only accommodate 25 particles. The particles originally in one layer must now be arranged on approximately four layers. During the course of flow, the particles must move relative to each other and rearrange themselves constantly, so that the mass of coal can be deformed to suit the decreasing sectional area of the hopper. In the meantime, the relative movement of particles must be accompanied with frictional resistance and requires the expenditure of energy to accomplish it. Obviously the greater the amount of deformation per unit distance of descent through hopper, the greater the amount of frictional resistance. This friction between particles depends on the following three factors:

1. The coefficient of friction between the particles.
2. The pressure between them.
3. The amount of deformation per unit distance of descent.

The last factor, deformation of coal body, depends on the rate of area reduction which is defined as the reduction of cross-sectional area per unit distance of descent with respect to the original area. The coal near the wall has more deformation than that at the center of the hopper. However, if the wall is fairly steep, it will not introduce much error to assume that the rate of area contraction is uniform through any section. Let A denote the original area and A' the area one foot below. Then the average rate of area contraction C may be expressed as:

$$C = \frac{A - A'}{A} \qquad (7)$$

or in mathematical expression

$$C = -\frac{dA}{dy} \Big/ A = -\frac{dA}{A\,dy} \qquad (8)$$

The rate of area contraction depends on the shape and dimension of the hopper and represents the amount of deformation per unit volume of material flowing through the section of hopper.

The resistance due to friction between particles is therefore expressed as:

$$Z_2 = Pf_2\, kC \qquad (9)$$

where

Z_2 = resistance per unit volume of material due to area contraction
f_2 = coefficient of friction between particles

k = a constant involving size, shape of particles.

The resistance of wall friction per unit volume of coal in the hopper could also be represented by equation (4) if the inclination of wall is fairly steep. Now the values of U and A are not constant as in the cylindrical hopper but vary with the elevation of the section. If the wall slope is relatively flat the resistance would be better expressed by

$$Z_1 = Pf_1\eta\ (U/A)\ CSC\ \varphi \qquad (10)$$

where φ is the angle of the wall with the horizontal.

The downward pressure in the hopper may be generally expressed as:

$$P = Wy - \int_0^y Pf_1\eta\ (U/A)\ dy - \int_0^y Pf_2kCdy \qquad (11)$$

Proper downward pressure is necessary to maintain continuous discharge of material. The flow is likely to be interrupted if the pressure reduces to zero. For simplicity, resistances and pressures on the same elevation in the hopper are assumed uniform in this equation since there are already so many variables that the general solution becomes overly complicated.

Why a Conventional Hopper Chokes

Conventional hoppers are usually made with walls of constant slope from top to bottom although the horizontal cross section may be either circular, square or rectangular, and the side elevation may be either symmetrical or asymmetrical. It is the inherent characteristic of this design that both the wall friction factor $[\eta(U/A)]$ and the rate of area contraction $[(A - A')/A]$ increase rapidly toward the discharge opening. Refer to Fig. 3-1, which represents a typical conventional hopper of square cross-section and 60-deg wall slope with an 18 sq in. discharge opening. As the material flows from elevation 13 to 12, the width (d) is reduced from 8.5 to 7.4 ft and the average rate of area contraction is 28 per cent per foot of descent. From elevation 8 to 7 however, the average rate of area contraction becomes 68 per cent. Fig. 3-2 shows the rate of area contraction increase in the direction of flow in the hopper. The high rate of area contraction causes more deformation of coal body and higher flow friction.

The wall friction, which is proportional to $\eta(U/A)$, becomes $4\eta d/d^2$ or $4\eta/d$ for square hoppers, η being constant since the wall slope is constant. From this expression, it is obvious that the wall friction increases as the width (d) decreases, as shown in Fig. 3-3. As discussed above, both particle friction and wall friction increase in the direction of flow and reach a maximum at the discharge opening. Furthermore, lower resistance in the upper section permits the build up of pressure which compacts the material at the lower end. When the total resistance exceeds the gravity force, the flow of material will be interrupted.

In the above discussion, both resistances are assumed uniform over the entire cross-section area. Actually more area contraction exists near the wall, especially in the corners, affecting the coal in contact with the wall. Therefore, certain material would let the central core right above the opening flow down while the remainder "hangs up" in the hopper due to additional resistance near the wall. Sometimes due to the presence of moisture or compaction, an undeformable coal body or "bridge" is formed over the opening and plugs the flow entirely.

Increasing of hopper wall steepness can relieve the difficulty to some extent, but does not solve the problem completely. A hopper with a 75-deg wall slope still has a rate of area contraction of 48 per cent per foot. This type proved better with free flowing material but sometimes caused more trouble with wet coal. Further increase of steepness does not seem economically justified.

Hyperbolic Hopper Outlet Studied

Since clogging of flow in the conventional hopper always originates at the discharge opening, a new type outlet has been designed to replace the bottleneck. This device, as shown in Fig. 4-1, has hyperbolically curved walls and is therefore called a "hyperbolic hopper outlet." It is based on the idea that gradual increase of wall steepness has the effect of limiting the rate of area contraction and reducing the wall friction. As its width is larger than the corresponding one of conventional design, the rate of area contraction is limited within

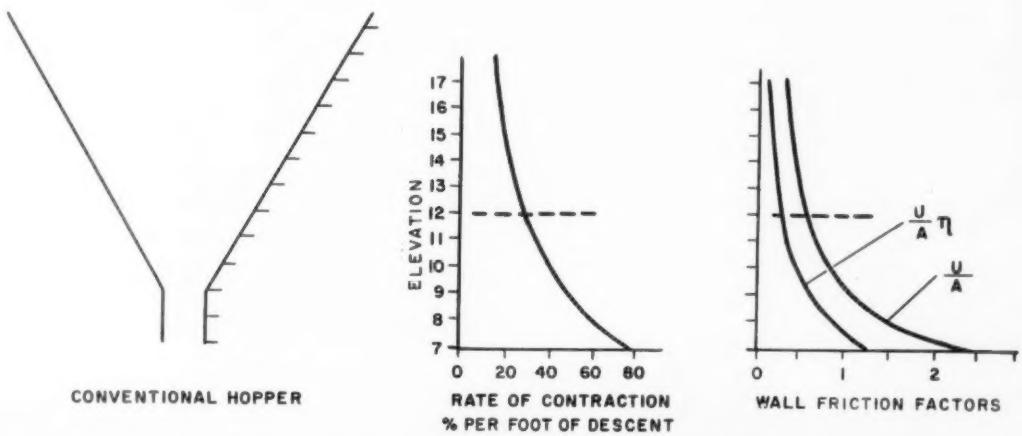


Fig. 3-1 Conventional hopper with elevations corresponding to Fig. 3-2, middle figure. This shows variation in rate of contraction and Fig. 3-3, right figure, shows variation of wall friction factors

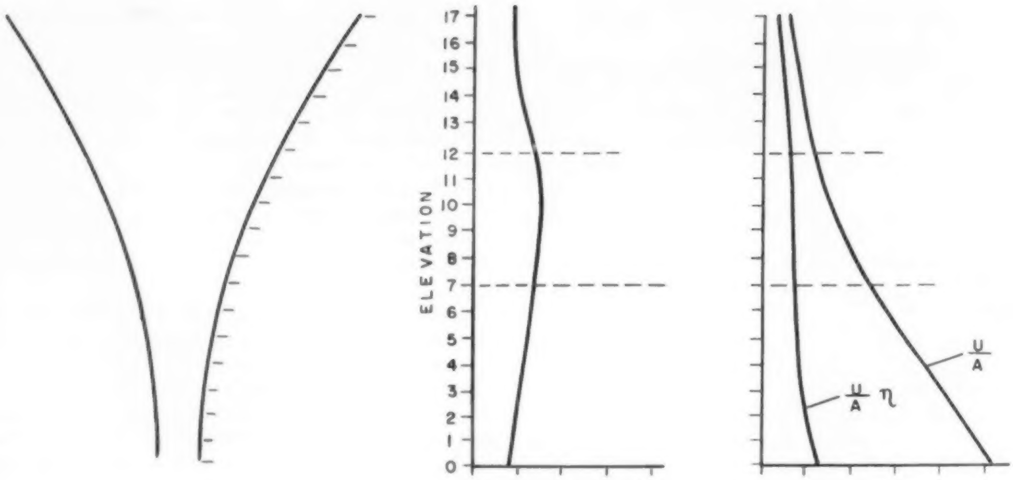


Fig. 4-1 Hyperbolic hopper with elevations corresponding to Fig. 4-2 showing variation in rate of contraction and Fig. 4-3 showing variation in wall friction factors

a reasonable value as shown in Fig. 4-2, which is obviously far less than that shown on Fig. 3-2. The amount of coal body deformation and the friction between particles are also maintained within reasonable limits.

Concerning the wall friction, it is found that the inclination factor decreases as the wall steepness increases. It is 0.55 with a 60-deg wall, 0.40 with a 70-deg wall, 0.30 with an 80-deg wall and 0.21 with a vertical wall. The gradual decrease of inclination factor will compensate for the increase of resistance due to the reduction of passage area to some extent. The final wall frictional factor for the hyperbolic hopper outlet is shown on Fig. 4-4 which is far less than that on Fig 4-3.

The curvature of the wall is vital to the success of the hyperbolic hopper. There are several ways to design it so as to fit individual conditions. One way to do it is to set the rate of area contraction constant so as to keep the friction between particles per unit volume of material also constant. This is applied if the decrease of inclination factor can compensate for the effect of area reduction. Then:

$$-\frac{dA}{Ady} = C = \text{constant} \tag{12}$$

$$\frac{dA}{A} = -C\,dy$$

$$\text{Log } A = Cy + k_1$$

$$A = (e^{k_1})\,(e^{-dy}) \tag{13}$$

At the juncture of hyperbolic hopper outlet with the upper hopper, $y = 0$, $A = A_0$, $d = d_0$, the above equation becomes

$$A = A_0 e^{-dy} \tag{14}$$

In the particular case of a square type hyperbolic hopper outlet having symmetrical walls, the profile is

$$\chi = \pm\,\frac{d_0}{Z}\,e^{-(c/z)y} \tag{15}$$

Sometimes it is found that the hyperbolic hopper outlet wall friction actually increases slightly near the discharge opening. In this case, it is preferable to have the rate of area contraction reduced slightly toward the opening so as to keep the resistance of flow constant along the passage. This may be determined, for example, through the utilization of the following expression:

$$-\frac{dA}{Ady} = \frac{2}{k_2 + y} \tag{16}$$

$$\frac{dA}{A} = -\frac{2}{k_2 + y}\,dy$$

$$A = \frac{A_0 k_2^2}{(k_2 + y)^2} \tag{17}$$

For square and symmetrical hoppers, the profile will be:

$$\chi = \pm\,\frac{d_0}{2}\,\frac{k_2}{k_2 + y} \tag{18}$$

Another way to design is to calculate the area or width for each foot of descent using the following:

$$A' = A(1 - C) \tag{19}$$

where C is the average rate of area contraction, A' is the area one foot below that of A. For a rectangular body having a width d_1 and d_2 and area, $A = d_1 \times d_2$:

$$d_1' = d_1\sqrt{1 - C} \tag{20}$$

$$d_2' = d_2\sqrt{1 - C} \tag{21}$$

The points calculated from the above are connected to form smooth curves and the slope of the curve at each

point is measured to determine the corresponding inclination factor η. The value of $\eta(U/A)$ is then calculated and if it exceeds that on the above section, the value C is reduced and d_1 and d_2 are recalculated.

As both particle friction and wall friction are much smaller, the combined resistance of the hyperbolic outlet should also be far less than that of the conventional one. The maximum resistance for the outlet indicated on Fig. 4-1 is only 40 per cent of that on Fig. 3-1, and only 66 per cent of the conventional design with a 75-deg wall. Therefore the coal can flow down steadily, uniformly and faster.

Furthermore, the pressure distribution in the hyperbolic hopper outline constitutes another advantage. The pressure equation (11) for this type hopper can be rewritten as:

$$P = Wy - \int_0^y P(f\eta(U/A) + f_2kC)dy \qquad (22)$$

Since the hyperbolic hopper is designed with the combined particle friction and wall friction constant or approximately constant, it follows that:

$$f_1\eta \frac{U}{A} + f_2kC = k = \text{constant} \qquad (23)$$

$$P = WyP\int_0^y Pkdy \qquad (24)$$

$$\frac{dP}{dy} = W - Pk$$

$$\frac{dP}{W - kP} = dy$$

$$-\frac{1}{k}\log(W - kP) = y + C \qquad (25)$$

Letting the pressure at the junction between the upper hopper and the hyperbolic hopper (where $y = 0$) be represented by P_0:

$$C = -\frac{1}{k}\log(W - kP_0)$$

$$\operatorname{Log}\frac{W - kP}{W - kP_0} = -ky$$

$$P = \frac{W}{k} + \left(P_0 - \frac{W}{k}\right)e^{-ky} \qquad (26)$$

From this equation, it is obvious that as y increases, $[P_0 - (W/k)]e^{-ky}$ diminishes and P approaches W/k. This pressure is rather uniform through the outlet, not high enough to cause compaction nor too small to maintain continuous flow. (See Fig. 4-4.)

Flow of Cohesive Coal in Hoppers

The above discussions are confined to cohesionless coal. When adhesive material is handled, the relations given above should be modified to include the effect of cohesion.

The cohesion of coal is mainly due to the presence of moisture in it. The water film on the material has a surface tension effect which tends to hold the particles together or to cause them to adhere to the walls. It

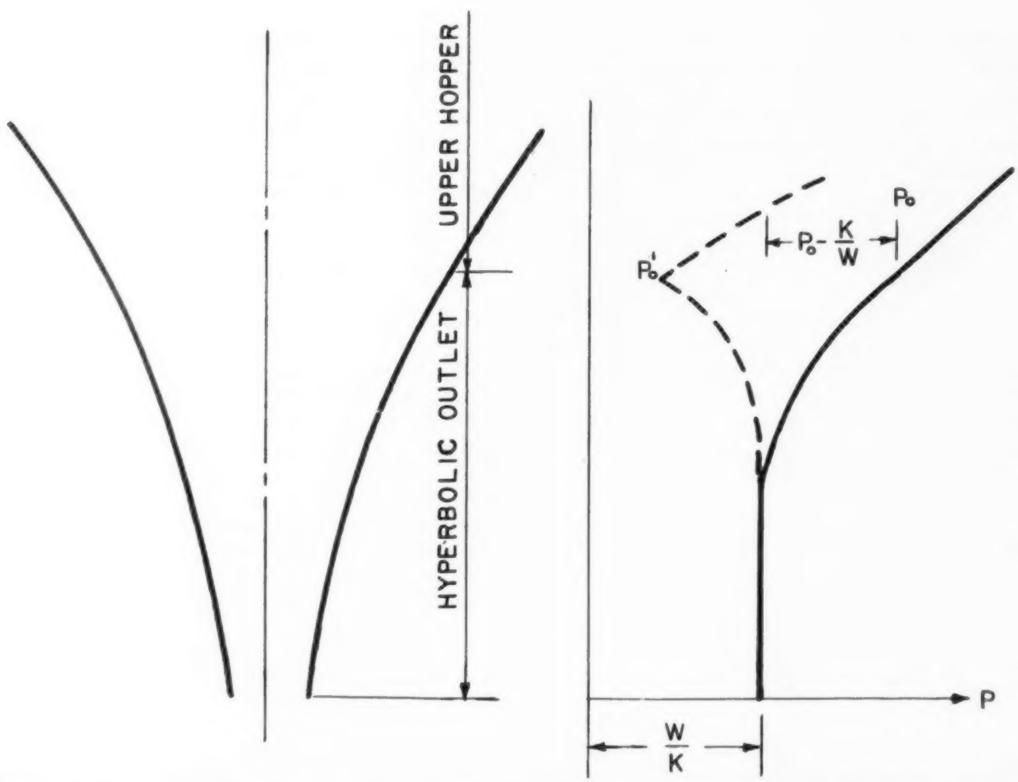


Fig. 4-4 Illustration of Equation (26) which expresses the relationship between coal pressure, (P) and the pressure (P_0) at the junction of the upper hopper and the hyperbolic hopper outlet

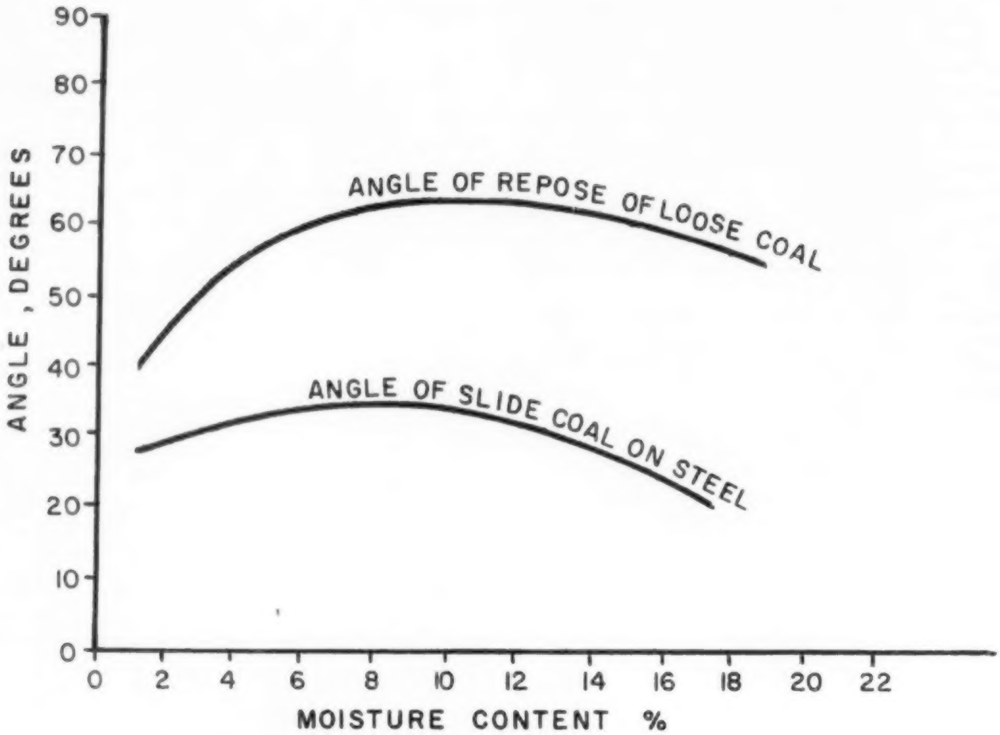


Fig. 5-1 Curves showing how the characteristics of one coal vary with moisture content

causes resistance to the flow as friction does, but differs from friction in that cohesive resistance is independent of pressure.

Let c_1 be the cohesive resistance with unit area of hopper wall and c_2 that between particles. Then the total wall resistance per unit of wall area will be:

$$R_1 = P\eta f_1 + c_1 \quad (27)$$

and the total unit resistance due to relative movement between particles:

$$R_2 = Pf_2 + c_2 \quad (28)$$

The friction factors f_1 and f_2 are practically independent of moisture while cohesions c_1 and c_2 depend on the moisture content. The variation of wall resistance and the resistance between particles may be represented by the angle of slide and angle of repose, respectively. Those angles differ with different coals. Fig. 5-1 shows the curves for one coal sample.

For cohesive coal, the equation of downward pressure becomes:

$$P = Wy - \int_0^y (Pf_1\eta + c_1)\,(U/A)\,dy - \int_0^y (Pf_2 + c_2)kCdy$$

$$= Wy - \int_0^y [c_1\,(U/A) + kc_2C]\,dy - \int_0^y P\,(f_1\eta(U/A)f_2kC)dy$$

This equation is the same as equation (11) with the exception of the second term which represents the cohesive effect. The solution of this equation is overly tedious but it is obvious that the downward pressure is reduced by the cohesive effect, represented by the second term. If P were reduced to zero, the coal would cease flowing.

In the case of the hyperbolic hopper outlet, where either $\eta(U/A)$ and C are constant or $[\eta(U/A) + k'C]$ = a constant, $[c_1(U/A) + c_2kC]$ would not vary very much. Assuming this is true; $[c_1\,(U/A) + c_2kC]$ may be represented by k' and the equation becomes:

$$P = Wy - k'y - \int_0^y Pkdy$$

$$= (W - k')y - \int_0^y kPdy \quad (30)$$

$$P = \frac{W - k'}{K} + \left(P_0 - \frac{W - k'}{K}\right)e^{-ky} \quad (31)$$

Comparison of this equation with (26) reveals that the density of material (W) is replaced by ($W - k'$). In other words, cohesion is the equivalent of a reduction in coal density. In order to keep proper downward pressure, it is necessary to keep the value of k' low and sufficiently constant. In the handling of cohesive coal, the hyperbolic hopper outlet with its reducing rate of area contraction must always be preferred.

Application of the Hyperbolic Hopper Outlet in Peipu Power Station

The hyperbolic hopper outlet was first put into experimental use in the Peipu Power Station of the Taiwan Power Company where serious coal flow trouble had existed for decades. Months of operation proved the effectiveness of the device.

The coal used in this station is low grade bituminous, $^3/_4 \times 0$ in. slack with approximately 48 per cent passing $^1/_8$-in. screen. As it is dug from narrow coal

seams, shale and clay are usually mixed with it. Furthermore, the coal absorbs large amounts of water during the prolonged rainy season, and for these reasons this coal is usually cohesive and troublesome.

The first hyperbolic hopper outlet was installed on a coal bridge for loading the reclaimed storage coal on a belt conveyor. As might be expected, the experimental hopper discharged coal uniformly and continuously even though it was not properly fabricated. This change permitted the release of the hopper attendant for other duties and allowed the conveyor to be loaded to its maximum capacity. A second hyperbolic outlet was fabricated to replace the conventional outlet of a 250-cu m boiler bunker. During a test period or two weeks, the hopper of original design, although equipped with electric vibrators encountered 353 interruptions while the remodeled one did not have a single interruption even though the same slack of 14.3 per cent surface moisture and 30 per cent ash was handled. It was observed that the coal in the hyperbolic section descended uniformly. The coal flow in the upper hopper is not limited to the central core alone but covers a large area of the bunker. The bunker volume could thus be utilized to its full capacity. After the test, all the hoppers in this plant were remodeled accordingly and the vibrators were removed for use elsewhere.

Since this change the coal flows by itself and no more pounding or poking is needed. The boiler operating force has been reduced by 13 men per day. The generating units can now operate continuously at maximum capacity and the average plant output has been increased by 5 per cent. Thermal efficiency of the plant has also improved due to steady fuel air ratio. Another important advantage occruing from these new hopper outlets is that the power output has become reliable no matter whether the coal is wet or dry.

Influence of Hyperbolic Hopper Outlet on Feeders

The hyperbolic hopper outlet was first developed in order to eliminate clogging of coal flow in hoppers. Recent investigation indicated that it helped the volumetric feeder to give more accurate volume control.

From tests we found that the density of bulk materials varies with the pressure they undergo. Taking the coal used in Peipu Power Station as example, the density is 0.76 in the dry, loose state. When it is compressed to 17 lbs/sq in. pressure, the density becomes 1.01. The variation of coal density at different pressures is shown on Fig. 6-1. The density variation of wet coal is still more significant.

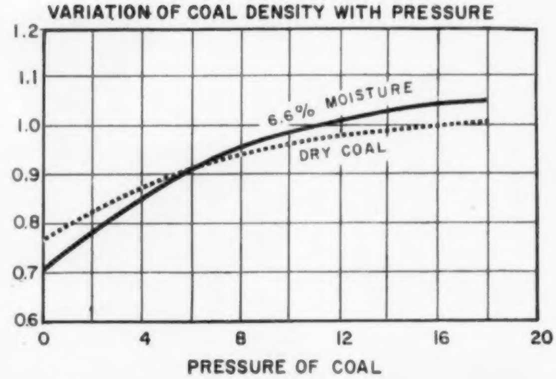


Fig. 6-1 Curves showing the variation of coal density with pressure

In volumetric feeders, the weight of material fed is equal to the product of density of the material, the volume of the vessel and feeding speed. Assuming that the density is constant, regulation of feeding speed will control the weight of coal fed. But since the density depends upon the pressure, the volumetric feeder gives accurate control only when the pressure of the material just before the feeder is constant.

In a conventional hopper, the pressure of bulk material at the discharge opening is not constant but varies with the following factors:

(1) the level of material in the hopper.

(2) status of flow.

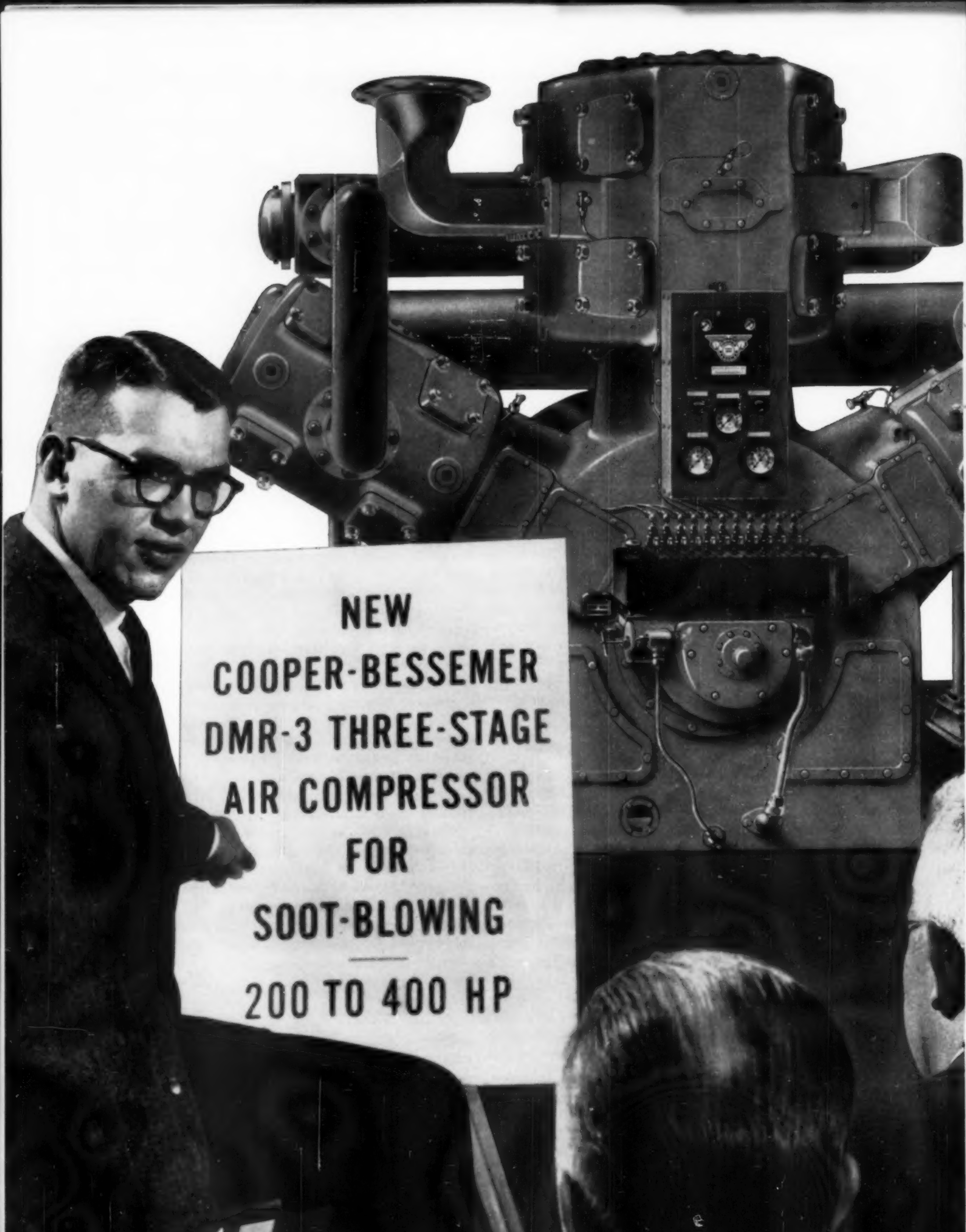
Andrew W. Jenike made a thorough study on conventional hoppers (*Chemical Engineering*, p. 175, Dec., 1954), and pointed out that the pressure at the discharge opening increases with the increase of level of material in the hopper. In case of funnel flow, the pressure fluctuates over a wide range. Therefore, because of these variations of pressure, the volumetric feeder cannot maintain a given desired rate of feeding.

Fortunately the hyperbolic hopper outlet, when mounted before the feeder, can compensate for this defect because as has been shown above, a properly designed hyperbolic hopper outlet will maintain constant pressure at the discharge opening. Experimentation is continuing in this aspect of coal flow and further progress toward constant feed rates maybe expected.

Engineering Institute Program

"We hope to bridge the ever-widening gap between operation and management (*Ed. Note—of electric utilities*) wherein the former is too often content to stand pat with today's level of understanding and the latter experiences more and more concern about staying abreast of advancing technology." In these words Paul Grogan, Chairman of the University of Wisconsin explains some of the objectives of the University's Engineering Institute Program on "Hydro and Electric Plant Operation." Specifically planned for utility plant operating and supervisory personnel, principles and employees of consulting engineering firms and for representatives of manufacturers and contractors of related equipment, the program will be presented in three, four-day meetings at the University of Wisconsin in February and March.

The three meetings are scheduled for Feb. 8 through 11, Feb. 29 through March 3 and March 21 through 24. All three meetings will cover essentially the same ground and have been designed at the request of, and to help fulfill the training needs of, utilities and industrial concerns. For more complete information readers should address Paul J. Grogan, University of Wisconsin, University Extension Div., Dept. of Engineering, Madison 6, Wisconsin.


NEW
COOPER-BESSEMER
DMR-3 THREE-STAGE
AIR COMPRESSOR
FOR
SOOT-BLOWING
200 TO 400 HP

Space Age Hydrostatic Test

Those who have counted the seemingly endless hours waiting for pressure to come up on a boiler undergoing hydrostatic testing will appreciate this article. Naturally those who have paid stand-by time to a full erection crew for the days required to hydro a new steam generator will also approve this fast and effective hydro-test method.

By R. A. NICKERSON*

THE progress of steam generator efficiency through more effective utilization of heating surface has wrought marked changes in the configuration of boilers. Controlled circulation has also contributed to the complexity of steam and water circuits within the modern boiler by permitting designers much greater latitude in arranging heating surface.

These modern configurations have created a knotty little problem when the time comes to put a hydrostatic test on a steam generator. This situation is particularly vexing and expensive in the case of new units. Erection crews must be paid to stand by and wait because a million lb-per-hr controlled circulation boiler, for instance, requires about 16,000 field welds and it would be unnatural not to have to repair a few leaks.

The conventional method of applying pressure for hydrotest is to use the boiler feed pump to the limit of its pressure and then to use a low capacity, high-pressure pump to reach test pressure. These small pumps used by most erection forces take an inordinate amount of time to raise test pressure because they are required to compress a large quantity of air trapped in the complex circuitry of the modern boiler.

In recent years a small vacuum pump has been successfully employed to remove this trapped air from the circuits with the happy result of drastically reducing the time required for hydrostatic testing. This method has a further advantage in that pinhole leaks that were formerly impossible to detect since they leaked air, now can be located. These tiny leaks if undetected will cause later boiler outages. The saving in dollars from prevention of such outages while impossible to calculate must be considered.

Following is a brief synopsis of hydrostatic test of Unit No. 5 at Boston Edison's Mystic Station. Unit No. 5, shown in Fig. 1, generates 935,000 lb per hr at 1940 psig.

EQUIPMENT	CONNECTED TO	COST
1 Motor-driven vacuum pump with 3-hp, 220-volt, 3-phase motor; with snubber and filter	Top of superheater	$740.00
To evacuate 750 cu ft to 28-in. Hg in approximately 3 hr		
1 Air-Driven pressure pump, 5000 psi, approximately 1 gpm	Top of drum	
1 Boiler feed pump, 2640 psi		

Telephone communication is needed between upper drum, L.T. superheater outlet header drain valve and boiler feed pump

PROCEDURE TO HYDRO MAIN SECTION OF BOILER AND H.T. SUPERHEATER

With upper drum vents, drain valve at lower inlet header of L. T. superheater, and drain valve at L.T. superheater outlet header open, start filling boiler at the normal rate using the boiler feed pump.

Fill until water shows at vent at top of upper drum. At this point close all top vents and call the operator of the boiler feed pump to reduce the filling rate, so that the L.T. superheater can be filled in approximately 20 minutes.

Water will not flow to the steam-cooled wall tubes. When water shows at drain value at lower inlet header of the L.T. superheater, close this value.

Water will now start filling the L.T. superheater. Station a man at the telephone located at the drain valve for L.T. superheater outlet header. When water shows at this valve stop the boiler feed pump immediately. Do not allow water to go over into the H.T. superheater.

Check all valves for tightness and start vacuum pump. Pull a vacuum of 26 to 28 in. and when this is reached, valve off the vacuum pump. Start the boiler feed pump and bring it up to its top pressure.

Valve off the boiler and start the H.P. test pump.

RESULTS OF HYDROSTATIC TEST OF STEAM GENERATOR

March 4, 1959

11:30 a.m.—Started filling boiler using boiler feed pump.

11:50 a.m.—Boiler feed pump shut down. Water level lowered to repair leak in water wall tube 6 ft above lower drum.

1:02 p.m.—Boiler feed pump started and filling of boiler resumed.

1:55 p.m.—Boiler filled so that water was showing at drain valve on L.T. superheater outlet header.

* Engineering Dept., Boston Edison Co.

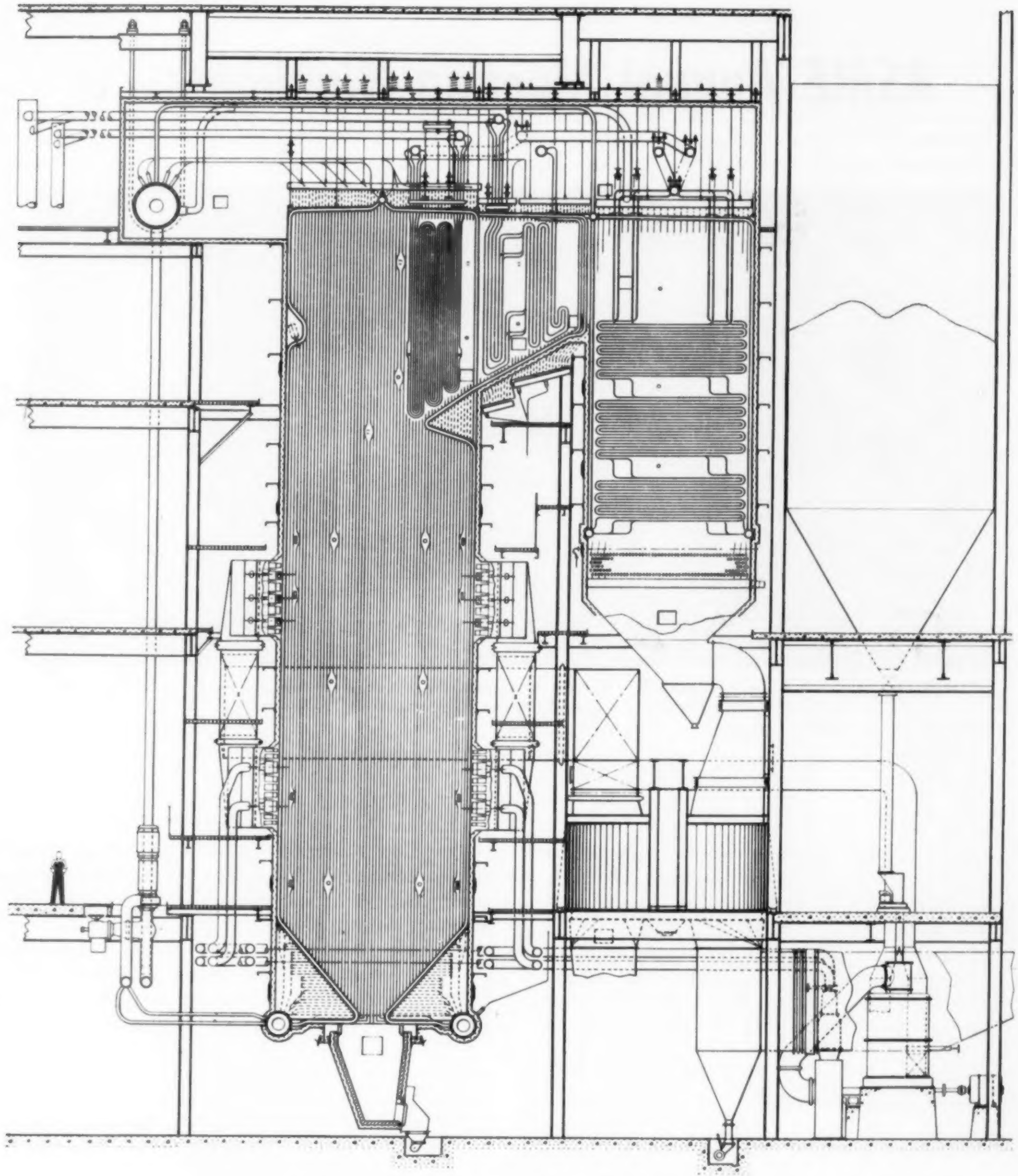


Fig. 1. Unit No. 5 at Mystic Station of the Boston Edison Co.

2:10 p.m.—Boiler valved off and vacuum pump started.
4:45 p.m.—Vacuum pump stopped at 26-in. Hg vacuum.
4:55 p.m.—Started boiler feed pump. Next hour used in filling, checking all valves and raising pressure to 600 psi.
5:50 p.m.—600 psig. Checked boiler for leaks from this pressure on.
6:50 p.m.—1700 psig on boiler.
7:12 p.m.—2000 psig on boiler
7:25 p.m.—2600 psig on boiler. This is maximum boiler feed pump discharge pressure. At this point eight leaks and weeps had developed. Boiler feed pump shut down.
7:36 p.m.—Started H.P. test pump.
8:30 p.m.—2840 psig on boiler.
9:35 p.m.—3225 psig or test pressure on boiler. Boiler inspected and leaks marked for repair.

PROCEDURE FOR HYDRO OF REHEATER

1. Install a 1-in. line between boiler feed pump discharge piping and a 1-in. connection on boiler side of reheat stop valve.
2. Blank off reheater at reheat inlet header.
3. Connect vacuum pump at reheat outlet stub header vent.
4. With all valves to or from reheater closed, start vacuum pump. When vacuum reaches 27 or 28 in., valve off vacuum pump.
5. Start boiler feed pump and fill reheater and (hot) reheat line.
6. Bring pressure up to 790 psig and check for leaks.

Inasmuch as the cost of the vacuum pump is negligible, the saving in time and dollars can be appreciable and the reduced possibility of future outages from pinhole leaks is attractive, the Boston Edison Co. has standardized on this approach to hydrostatic testing.

ASME Annual Meeting Highlights—II

Walker L. Cisler addresses ASME members at a banquet during the Society's Annual Meeting at Atlantic City in December. In the foreground is Glenn B. Warren, outgoing president of ASME.

IN this issue we conclude our presentation of abstracts from papers presented at the Annual Meeting of ASME. Since this was one of the Society's most successful meetings with a registration close to 4000 and over 300 papers it was obviously impossible to abstract all of the papers presented. We have tried to select some of wide general interest and since "Eddystone" was tantamount to a password at Atlantic City we have gathered immediately below those papers that tell the Eddystone story. Following them are other excellent papers given at this same meeting.

"The Eddystone Research Story" by **E. C. Chapman** and **R. E. Lorentz, Jr.,** Combustion Engineering, Inc., while not the first of the many papers on this subject presented at the meeting so well pinpoints the background for these Eddystone sessions we run it as the lead story.

The Eddystone supercritical boiler, designed to operate at a maximum pressure of 5000 psig and 1200 F temperature, required the use of new high-temperature materials in tubes and stop valves in order to avoid excessive wall thicknesses. None of the materials approved by the ASME Boiler and Pressure Vessel Code for superheater service had the required properties. The paper describes the high-temperature tests carried out and the welding development which led to the selection of the new materials. The properties of these new materials are discussed. Design features and fabricating procedures for the tubes and valves are presented.

The chief deterrent to the development and utilization of new materials for high-temperature service is the time required for carrying out the necessary long-time creep and rupture tests. Costly alloys of the austenitic-steel variety must give many years of service to make their use economical for service in steam generation. In contrast with aircraft-engine service requirements for which materials can be tested for the full life of a unit, the life of parts under creep in boilers must be predicted by extrapolating data derived from tests extending over only a fraction of the required life of the material while this is unfortunate it has not materially hindered progress.

17-40 CuMo steel, it was found, has high-temperature strength superior to the strength of ASME approved austenitic steels, and tubes of this analysis can be made using conventional steel-mill procedures. Tubes of this alloy can be fabricated without difficulty provided close control of heat-treatment is exercised.

Type 316 is a very satisfactory alloy for fabricating and has high-temperature properties which seem to testify to present stress code values.

Jessop G-18B alloy is an excellent material for many applications and deserves consideration as a material for use at the higher temperatures not only for valves but possibly for tubes and other parts.

The Fox CN-16/13 Co electrode appears to possess characteristic and provide high-temperature properties in deposited metal superior to any known American welding electrode. Satisfactory welds, completely austenitic, are possible and this is an important and unusual achievement.

R. H. Caughey and **W. G. Benz, Jr.,** M. W. Kellogg Co., collaborated on the paper "Material Selection and Fabrication Main Steam Piping Eddystone No. 1 1200-F and 5000-Psi Service." In the course of making the critical review of materials which led to the selection of the Type 316 composition for the Eddystone main steam piping it was found that, with only a few exceptions, there was a dearth of information on the weldability and weld joint properties of the austenitic stainless steels which might be relied upon to reflect the possible effect of mass or thickness contemplated in the piping design, of nearly 3 in. The effects due to thickness which might be related to the forging and thermal history in both steel production and pipe manufacturing and, in welding fabrication, the high degree of restraint which it is known increases in severity with increased thickness of the weld joint, were appropriately questioned. Therefore, the materials evaluation program was designed to feature the welding and testing of weld joints completed in full-thickness pipe sections comparable to the Eddystone pipe dimensions.

A suitably ground and lightly acid-etched, full-thickness cross-section of each weld joint was explored by the Rockwell hardness method employing the "B" scale of hardness. Values for the base material, weld deposit and heat affected zone in each case were recorded. The average of the values for each location in each weld joint in each condition were shown in table form. A summary of these values with respect to the separate component materials was also shown.

Generally, the hardness levels are within the expected range for the individual material compositions. The only noticeable change in hardness was in the Type 316 and 16-8-2 weld metals, both of which softened somewhat as the result of the 1950 F solution heat treatment. Aging, up to 10,000 hr at 1300 F, did not produce any marked change in the relative hardness of any weld joint components.

Many other factors of material selection were reviewed by the authors in completing this paper.

"Research Problems Relating to Production and Quality Control of Ultra-pure Feedwater for Eddystone Station" was discussed by **J. A. Levandusky** and **V. J. Calise,** Graver Water Conditioning Co. The design of the supercritical boiler has forced the creation of a new criterion for determining water purity for this type of plant. As the authors stated we now talk of ultra-pure water where the contaminants are measured in parts per billion and we differentiate between dissolved and suspended solids, since these solids have different effects on the water-treatment system; consequently the water treatment must be designed to deal with suspended as well as dissolved solids.

There are three sources of contamination in the water cycle: (1) dissolved and suspended solids in the makeup cycle, (2) leakage of dissolved and suspended solids from the condenser, and (3) pickup of metallic oxides such as iron, copper, chromium, molybdenum, and zinc by virtue of contact of steam and water with the metals in the cycle. To reduce the dissolved solids in the makeup water a makeup demineralizing plant is provided.

The condensate-purification system will provide for the reduction of the suspended metallic oxides that result from pickup by virtue of contact of the steam and water with metal surfaces in the cycle and the reduction of the dissolved solids, including silica, that enter the condenser stream from condenser leakage and weepage. The reduction of these solids will be to less than 10 ppb of suspended solids and less than 50 ppb of total dissolved solids, including silica.

In concluding the authors felt they must say that the true test of operation of the condensate-purification system will be made when the Eddystone Station goes into operation. Based on the pilot-plant data they have confirmed the ability of the equipment selected to produce the ultra-pure water required for the Eddystone Station and have obtained basic information on the operating characteristics of the precoat filters and ion-exchange equipment which will permit the operation of this equipment in the most efficient manner.

Also as a result of their test work they believe they have: (1) provided for filtration as an important part of the purification cycle for the removal of suspended solids; (2) established that iron and copper compounds will be reduced to the desirable levels and this is accomplished by filtration, not ion exchange; (3) demonstrated that in the event of a condenser failure the mixed-bed units will remove the solids and silica effectively; (4) extremely accurate analytical methods were worked out for plant control, and a new analytical instrument, The Tape Analyzer, will be used to aid in monitoring the system so as to insure that ultra-pure water is obtained at all times.

Somewhat along the same lines as the Calise-Levendusky paper the report "Pilot-Plant Water Studies for Eddystone Station" by **F. N. Megahan,** Philadelphia Electric Co., delved into experiences at other demineralizer installations on the Philadelphia Electric Co. system which had demonstrated the value of pilot-plant studies prior to actual operation of large scale plants. Only data gathered from pilot plant work on the actual site of the proposed plant, using the same water supply could be correlated to actual plant operation data.

Tests were conducted on water samples from 90 to 200 ft in depth, which were obtained while taking borings at the Eddystone site, to pursue the possibility of using well water as a raw-water supply. These samples contained approximately 250 ppm dissolved solids. Two city water supplies were available, each with solids of approximately 110 ppm. It was decided to use the city supply most readily available. This water originates in the Crum Creek. Daily analyses were run on the water and the solids proved to run very consistent over a 2-year period. Monthly averages of the total dissolved solids were plotted.

As a result of the findings during these 2 years of operation of the Eddystone pilot plant, a two-bed, mixed-bed system was installed at Eddystone Station using $8^1/_2$ per cent cross-linked cation resin and nonporous Type I anion resin in all units. This plant was in operation several months before No. I went into operation and has ever since been providing make-up water of the same high quality that was previously obtained from the pilot plant.

R. C. Ulmer, H. A. Grabowski and **R. C. Patterson,** Combustion Engineering, Inc., teamed up to present "Water Treatment, Corrosion, and Internal-Deposit Studies for Eddystone." Since, in the authors' opinion, water conditions could well decide whether the 5000-psi 1200-F Eddystone unit could be operated satisfactorily, every condition which might lead to a problem was given careful consideration. Tests run at supercritical pressure indicated that a considerable amount of "soluble salts" could be deposited in the 850- to 950-F range when initially present at 2.0 ppm. Less deposition occurred with potassium salts than with dodium, but it was sufficient to make careful control of solids in the whole system imperative. Demineralization of the make-up and bypass demineralization of condensate as required should give the desired control of total solids.

Further the authors' studies indicated that when present in the amount anticipated for Eddystone, even at 1200 F there was no measurable decomposition or deposition of salts. As to corrosion, proper control of dissolved oxygen and maintaining 10-ppb hydrazine and sufficient ammonia to give an 8.5 to 9.5 pH range, insured low metal pickup. Thus, deposition of metallic oxides in the boiler and turbine does not appear to be a problem. Corrosion tests and metallographic examination of the various alloys used indicate no harmful loss of metal or change in structure, hence no sercice-life problems are expected.

The tendency of an ammonia-hydrazine treatment to reduce the corrosion rate at high pressures and temperatures cannot at the present time be explained satisfactorily. Normally, ammonia separates from the liquid phase and passes off as a gas. The authors' present concept of the action may not be totally correct. Recent tests of the monotube unit at Dayton indicated an appreciable quantity of ammonia in the water removed from the separator. It is possible that ammonia, at elevated pressures, may remain in the homogeneous mixture passing through the unit without a complete separation from the dense medium.

As indicated, operation of the simulated boiler and turbine units has supplied much useful information. Especially it confirms design, selections of materials and planned water treatment and control.

In addition the acid wash of the test unit at the authors' Kreisinger Development Lab for the removal of soluble deposits have assisted in locating areas where deposits can form and can be of potential use to the designer in locating and proportioning the heat-transfer areas.

George C. Wiedersum, Jr., Philadelphia Electric Co., chose for his paper "A Test to Anticipate the Performance of Steam Piping at 5000 psi and 1200 F." As the author pointed out the Eddystone No. 1 Unit will operate under steam conditions of 5000 psi and 1020 F, and the effect of these extremes of pressure and temperature, especially the latter, is, to some extent, unknown. There are techniques available for predicting the effects of long-term exposure of metals at high temperatures, and in general their use has been successful in the past. These include the use of small specimens, loaded uniaxially, exposed at temperatures higher than the working point in order to obtain acceleration of the comparatively short periods of exposure. In the design of the components of this unit, use was made of all such methods possible, and the decisions reached were based on the best information available at the time.

The background of the decision to use Type 316 stainless steel for 200 tons of piping between boiler and turbine has been described in a recent paper by Blumberg. The development of the fabrication techniques was described by Caughey and Benz in the paper reported above. However, experience in the use of Type 316 in central station service is extremely limited; it has been used in only four other plants, the oldest of which went into service in 1957. The steam temperature in these plants is 1100 F, and the pressures are much lower than 5000 psi, so that the wall thicknesses are much less. The bulk of the usage of the material has been in petroleum refining, where its resistance to corrosion was its outstanding attribute, and temperatures were usually below 1000 F.

The use of Type 347 stainless steel in central stations is much more extensive, having been used in over twenty plants dating back to 1948. Steam temperatures have usually been 1050 F, with a few at 1100 F, and one at 1150 F. While generally succssful, there has been some circumferential cracking at welds with this material. The paper by Fairchild has recounted the difficulties experienced by one utility with Type 347. It is believed that Type 316, especially with the 16-8-2 electrode and 1900 F post-welding-heat-treatment used at Eddystone, is much less susceptible to cracking than Type 347, but this experience is cited to illustrate that there still are unknown factors involved in the use of austenitic stainless steels.

In view of the uncertainties, it was decided to develop a test that would predict as closely as possible the "in-service" performance of the Eddystone No. 1 main steam piping. Even if the test results were not obtained before the unit went into service, it would still permit taking some remedial action in the plant to forestall any difficulty that might be revealed in the test.

The author stressed that he realized this test has its limitations. Probably the most serious one is the use of only one sample. Statistically, the confidence limits of a single sample test are zero. However, the availability of material from original heats, and the expense of the samples and test apparatus preclude the use of more samples. Further, the authors believed, it is felt that making the test weld under difficult conditions and the slightly accelerated conditions of exposure insure that if anything is going to occur in the plant, it will occur in the test first, even if the probability cannot be calculated. Also, the companion testing done on the small samples lends a higher degree of statistical reliability to the overall picture.

Another possible shortcoming, the author suggested, is the weekly cycling period. This may not accumulate a large enough number of cycles in a reasonable length of time. Because of the inflexible number of hours that must be used to increase and decrease temperature, the weekly cycle was selected to produce a reasonable amount of time at the elevated temperature. If evidence is obtained that one factor, such as the cycling, is more important than others, the schedule will be revised accordingly.

In spite of these limitations, it is felt that the test will

provide the desired information on the performance of Type 316 stainless steel piping in 1200 F service.

W. E. Trumpler, Jr., E. A. Fox, A. F. LeBreton and **R. B. Williamson** all of the Large Turbine Engineering Dept., Westinghouse Electric Corp., reported on the "Development Associated with the Superpressure Turbine for Eddystone Station Unit No. 1". General design aspects of the 325-Mw Eddystone Station steam-turbine generator unit No. 1 were rather completely described in a paper presented at the 1956 Annual Meeting of the ASME. This machine was brought to completion during the intervening three years without major departure from those basic characteristics discussed previously. This paper supplements the 1956 report but limits its scope to some of the unique features of the superpressure element only.

This single element will be recognized by the relatively small component of the whole machine. Yet this element by receiving steam at 5000 psi, 1200 F, and expanding it to 2500 psi pressure at its exhaust, develops approximately $^1/_8$ of the entire rated output of the cross-compound double-reheat, turbine-generator set. Thus the superpressure element presented unique problems with regard to the selection and procurement of alloys suitable for the high service temperatures, as well as the development of efficient shaft and seals which have to operate against unusually large pressure differentials while occupying relatively little space.

The seals used reduce the axial space requirement for sealing by a factor of four when compared to previous radial seal labyrinth designs for the same leakage level. This has been accomplished by radially compounding the throttlings requiring the steam to follow a tortuous leakage path outward and inward radially as it moves axially along the shaft. There has been successful experience elsewhere with this general type of construction.

The logic of the use of radial, nested-type seals is obvious from a comparison of the shaft length required versus that to do an equivalent job with conventional seals. The bearing span of this particular superpressure-turbine rotor has been reduced approximately $3^1/_2$ ft by such change in construction along. The correspondingly stiffer rotor is of course less sensitive to minor unbalanced forces; there is substantially less liability to vibration at an amplitude such as to contact and rub radial seals, with resulting increase in steam leakage losses.

The seals at the end of the inner casing between the first-stage pressure and exhaust pressure are made slightly larger in diameter than those at the two ends of the outer casing. Therefore they form a piston to balance axial thrust resulting from the flow of steam through working stages of the superpressure element. The pressure drop across one rotating ring from the smallest to the largest diameter and back to the smallest diameter is not a linear function of the radial travel; therefore, the effective thrust pressure acting on the ring annulus is not the average pressure between inlet and outlet. To solve this problem analytical studies have been made of pressure distribution and factors have been developed for calculating thrust when the pressure ratio and the radii of the sales are known. By intentionally creating heavy radial and axial rubs it was found that this type of construction did not produce shaft bowing, always a possibility with conventional seals.

The principal factors considered in the selection of an alloy for the superpressure rotor shown were strength at room temperature, strength and metallurgical stability at operating temperature, thermal expansivity, availability and manufacturing and service experience.

D. Robertson, Leeds and Northrup Co., described "A New Sensitive Temperature Detector for Use in High Pressure Fluid Piping." Supercritical boilers, like Eddystone, make the conventional method of measuring steam temperatures within the piping unsatisfactory because of the severe stress environment. Wells, designed to withstand the environment, would have such massive proportions that they would be unable to follow temperature changes with the required speed.

A new method employing a special composite detector, is now being used for obtaining temperature measurements in well-insulated, heavy-walled piping with a speed of response equal to, or greater than, that experienced on low-pressure piping using conventional wells. The measurement is made with virtually no protrusion into the stream. Experimental test results, both in the laboratory and in the field, have indicated that the method is both practical and convenient to apply.

Mr. Robertson explained the new method as follows: "If the temperature of the inner surface of a heavy-walled well-insulated pipe (such as a steam line) should suddenly change and remain at the new temperature, the change will be felt progressively through the wall section from the inner to the outer surfaces. If a curve of temperature versus time was plotted for each of two points at different distances from the outside of the pipe, the two curves would start from the same point, diverge until a maximum temperature difference was reached and then would converge and finally meet when equilibrium was achieved again. The time required to reach a maximum deviation between the curves, is a function of the size and material of the pipe and the location of the temperature points, but is independent of the magnitude of the temperature change.

One way of utilizing the temperature-difference signal is by means of a special thermocouple. To make use of the temperature-difference within the wall effectively, the detecting system must be designed for the geometry and material of the pipe. For a particular internal temperature-change condition, a pipe with a 3-in.-thick wall will have a greater temperature difference per equivalent distance from the inner surface than a 1-in.-wall pipe of similar material and inside diameter. Likewise, the temperature difference in walls of pipes having equivalent geometry will vary inversely with the thermal conductivity of the pipe.

The ideal application of this system is to design the detector so that the temperature-difference signal is sufficient to produce a curve that becomes tangent to the final temperature line in a minimum of time, but is not sufficient to overshoot it. The temperature gradient between the inside and outside surfaces of the Eddystone line, under steady-state operating conditions, is expected to be about 4 F. Since the spacing between the couple junctions is expected to be about one-fifth of the wall thickness, the error should be less than 1 F.

Although only one type of detector has been described, other forms, Mr. Robertson reported, can be used. For example, if more temperature-difference signal is required and it is impractical to increase the spacing, the difference 'couple can be made of alloys having higher thermal EMF than the instrument calibration.

Alexander Brkich and **Robert E. Allen**, Ingersoll-Rand Co., joined in the paper "Development of Floating Ring Type Stuffing Boxes for Eddystone Boiler Feed Pumps." The continuous increase in power plant size during the past decade along with corresponding higher pressures and temperatures, brought new problems to boiler-feed-pump manufacturers, particularly at the pump stuffing boxes. Most previous experience was with 3600-rpm, 3 to 5-in.-diam shaft sleeves, suction pressures to 300 psig, and suction temperatures to 300 F. These conditions were handled adequately with conventional solid rings of packing. Now, however, the increased requirements necessitated a major departure from past practices. The floating ring design was evolved as the best approach to the problem.

Individual floating rings are locked against rotation by lugs and they seal against the adjoining stationary retainer with their raised, lapped faces and throttle leakage between their inner, annular surfaces and the rotating shaft. The individual floating rings are free to move radially in their retainers during momentary load changes and resulting shaft movements, always effectively throttling leakage due to their close symmetric clearance with the shaft sleeve. Springs hold the rings shut against the retainers when the pump is idle and the pressure is off. During operation, however, the pump pressure holds the rings shut while running. It is important to have sufficient injection pressure at all loads at a low enough temperature to prevent the hot water from flashing as it throttles underneath each ring.

When the Eddystone job came along there was no ready source of pressure and temperature combinations at its level but the Philadelphia Electric Cromby Station did have 2000 psig at 521 F and 2260 psig at 360 F available, so it was decided to build a tester which was actually a barrel pump without internal parts but having identical bearing, shaft sleeve and so on, as the proposed unit with individual floating-ring bundles adjusted in length to obtain the same pressure drop per ring as expected in the final high-pressure-pump, stuffing-box layout.

This investigation brought out the fact that a controlled leakage seal must accomplish two things: (1) it must reduce the pressure in a suitable manner, and (2), it must also cool the shaft so that at no place is its surface temperature greater than the vapor temperature of the water in the vicinity. This means that seals can be shortened by using very high pressure drops only if the proper cooling is taking place.

Testing continued up until the time that the Eddystone pumps were actually built in order to take advantage of any new results that may be uncovered. The entire program took 2 years to complete and during that time, over 2800 hr of actual running time was logged. One point that became obvious was that there is no substitute for full-scale testing with work of this type. It was a fortunate decision to have the tester duplicate the final pumps as much as possible for it became clear after a while that the physical dimensions and the loadings were of paramount importance. Results obtained from smaller sized rings would have been completely misleading.

Corrosion

A further paper "Oil-Ash Corrosion of Superheater Alloys in a Pilot-Scale Furnace. . .Reduction by Use of Additives" was presented by **Norman D. Phillips** and **Charles L. Wagoner,** Babcock & Wilcox Co. The corrosion of metals by the oil-ash constituents vanadium, sodium and sulfur depends upon the melting characteristics of the ash. Below the melting temperature corrosion is generally negligible, but considerable attack occurs with liquid slag. Vanadium pentoxide melts at about 1202–1274 F, but when vanaduum combines with sodium in boiler atmospheres, low-melting sodium vanadates are formed. Melting temperatures average about 1150 F but can vary, depending upon the ash composition, with some melting temperatures as low as 1000 F having been obtained.

The published corrosion data have been obtained mostly with isothermal test conditions applicable to gas-turbine operation in which the temperatures of the combustion gases and the metal are nearly equal. In boiler operation, however, the two temperatures are different, and these differences have a significant effect on corrosion rates. Because of these temperature differences we have not been able to transpose directly the gas-turbine data.

In order to investigate oil-ash corrosion under conditions where the gas temperatures and metal temperatures are different, and to determine methods of alleviating corrosion, a research program has been in progress at the Research Center of the authors' company.

This paper described the tests that were performed to study the effect of fuel-oil composition and metal and gas temperature on corrosion of selected high-temperature alloys. It also presented data on the effectiveness of additives in reducing the severity of such metal corrosion.

These tests showed that none of the four alloys tested was outstanding in its corrosion resistance; however, Type 310 indicated some promise at the higher temperatures. The relationship for a specific fuel-oil composition was shown for the combined corrosion effect of metal temperature and gas tempreature. With this type of relationship the authors believe they shall be better able to predict metal loss at various sections in a boiler. Also, after obtaining similar data for other fuel compositions, they shall be able to translate previous isothermal data to a combination of temperature conditions.

Novanadium type fuels of less than 10 ppm V produce much less corrosion than do oils of 150 ppm or more. The amount of sodium does not have as significant an influence on corrosion as does vanadium. Additive materials fed as a dry powder produced sufficient reduction in corrosion so that alloys at 1200 F metal temperatures with additive suffered less corrosion than they did at 1100 F without additive. Optimum additive rates of at least three times the vanadium content as the additive cation, are required with 1200 F metal temperatures over a range of gas temperatures between 1400–2400 F.

Fuel Additive

R. J. Zoschak and **R. W. Bryers,** Foster Wheeler Corp., reported on "An Experimental Investigation of Fuel Additives in a Supercharged Boiler." This experiment took the form of a test program wherein a number of magnesium and aluminum-bearing additives injected into washed residual oil when firing a laboratory-scale, simulated supercharged boiler. Different tube arrangements within the boiler were tried. Ash collected on the tubes at various locations was analyzed and its corrosive effect at high temperatures on some types of stainless steel evaluated.

Based on the test results obtained thus far on the test rig, the authors drew the following conclusions: (1) When magnesium is the sole cation in the additive, the amount and nature of deposits are essentially unchanged regardless of whether the additive is oil-soluble, a water solution or a suspension of solid particles. (2) The highest ash-fusion temperature attainable using magnesium as an additive is in the vicinity of 2400 F. (3) Soot blowing is relatively ineffective at the elevated gas temperatures existing in the test unit. (4) The composite, inhibited ash from any given location is not corrosive to the alloy steels tested at temperatures up to 1600 F when the additive ratio is an low as 1.5 Mg : 1 V.

In addition the authors advanced the following hypotheses based on evidence from these tests together with data of other investigators: (1) Ash deposits are being initiated by the diffusion and condensation of volatile vanadium pentoxide in the high and low-temperature gas zones and silica sulfide in the high-temperature zones. (2) Vanadium pentoxide is reacting with magnesium oxide and other minor constituents in the vicinity of the tubes. This may apply to the high temperature (1900 F +) gas stream only. (3) Sulfur trioxide is formed in the vicinity of the tubes and reacts with the ash constituents upon deposit. (4) When alumina and magnesia are present, magnesium aluminate is formed in the vicinity of the flame. No evidence of an alumina vanadate has been observed. The condition of sufficient magnesium to react with the aluminum may have a considerable effect on the role aluminum plays in the ash-deposition problem. (5) A solid solution of magnesium orthovanadate and magnesium oxide is formed when magnesium is added to the oil in the ratio of 1.5 magnesium to 1 vanadium or higher.

Low Level Economizer

J. H. Potter, Stevens Institute of Technology, and **R. C. King,** Gibbs and Hill, Inc., continued this subject treatment with the paper "Design Performance of the Low-Level Economizer." In this paper the possible improvement in economy resulting from a recovery of a larger portion of the flue-gas energy is investigated. Paramount in this study is the realization that the energy which is recovered from stack gases must be returned to the cycle. No gain would result from a reduction in stack temperature, for example, by exchanging heat with the atmosphere alone. Several schemes for the utilization of recovered energy are possible, among them being: (1) a transfer of heat, either direct or indirect, from the hot flue gases to the combustion air, (2) a transfer of heat from hot flue gases to the feedwater of the main power cycle, (3) a combination of the foregoing. (4) a system in which gases would be scrubbed by water. The energy recovered would be transferred to the main-cycle feedwater system, to protect the air heater by tempering, and to reduce the dust content in the stack effluent. (5) A system that would exchange heat from the hot gases to the feedwater system, but which would operate at condensation temperatures so as to recover water vapor from the products of combustion.

An existing 100,000-kw installation was selected as the basis for the present study. The plant is installed at an altitude 5000 ft above sea level and the steam-generating unit is arranged to burn coal, oil or natural gas.

The combustion air tempering scheme utilizing steam extracted from the turbine was selected arbitrarily as the one with which the low-level economizer installation is compared. The low-level economizer was selected to match the regenerative air heater furnished as a part of the steam-generating unit. That is, no attempt was made to select the most economical combination of low-level economizer and air heater. A larger and more expensive economizer would permit the use of a smaller and less expensive air heater. The optimization of these two pieces of equipment was considered to be beyond the scope of this present paper.

Pertinent performance characteristics of the economizer and tempering coil installation were shown graphically. Two basic requirements had to be satisfied by the installed equipment: (1) The air heater must be protected by means of supplying air at a sufficiently high temperature to match the minimum cold-end requirement; (2) the economizer must be supplied with water at a sufficiently high temperature so that the base of the extended surface will be safely above the water dew-point. The air heater manufacturer specified 210 F as the minimum cold-end temperature; the economizer manufacturer specified 100 F as the minimum water temperature at economizer inlet.

The uncontrolled economizer inlet water temperature proved to be 100 F at 16 F ambient air temperature. Thus, at full load at 16 F ambient and below, it was necessary to institute some sort of control for economizer protection. This embodied two features—an automatic temperature-controlled bypass valve and a supplementary steam heater.

Test results showed in addition to the savings in heat rate, for constant throttle flow, there was also considerable increase in net electrical generation due to the economizer installation. This increase in net output was greatest at low ambients which called for the the greatest quantities of bled steam; at high ambients, the increase became smaller and, at very high ambients, became negative due to two factors. First the amount of bled steam required is smaller, and second, the increase in frictional resistance and decrease in theoretical stack draft more than offset the extra energy due to the higher temperature of the combustion air.

Any economic evaluation of the relative merits of steam tempering versus low-level economizer tempering, the authors stressed, must take into account the loading schedule of the station, the differential heat rates, the variations in net station output and, of course, the installed costs of equipment.

Editor's Note: This paper proved so interesting Combustion will print it in its entirety in the February issue

The problem of whether or not a pumping installation and its supply and reserve are adequate to meet varying suction demands under normal cycle transients—COMBUSTION, May 1959, Steam Power Plant Clinic—VIII and COMBUSTION, November 1959, Steam Power Plant Clinic—XIII—has evoked so much interest and correspondence we are presenting a third discussion on this subject.

Steam Power Plant Clinic—Part XIV

Further Simplifications in the Analysis of Transient Conditions in Open Feedwater Cycles

By IGOR J. KARASSIK*

Worthington Corp.

Question

We continue to receive a considerable amount of correspondence in connection with the Steam Power Plant Clinics and mostly with regards to the material on "Transient Conditions." This subject is apparently one that is as close to the interests of the power plant operators as it is dear to our own heart.

The gist of a number of questions that have arisen has been a wish for a further simplification of the analysis which indicates whether or not an installation is adequate from the point of view of suction conditions. Some two or three years ago we had worked out such a simplification, although frankly speaking even the method we had developed previously and which is now used by many utility engineers is most simple and direct. On the other hand, this latest method does bring an important concept, that of the direct relation which exists between the size of the deaerator storage tank and the volume of the suction piping.

Answer

Of all the difficulties which may beset boiler feed pumps operating in large central stations, the most insidious perhaps are those caused by the transient conditions which occur during sudden load reductions in open feedwater cycles where the boiler feed pump takes its suction from a deaerating heater floating on an extraction stage of the main turbine. Frankly speaking, this is a subject which has fascinated me for a number of years with the result that I have devoted considerable time and effort to its study and to the development of methods of calculation which would indicate very readily whether a projected installation will or will not give rise to serious operating problems. These studies, in turn, have resulted in the publication of a number of articles on this subject in the technical literature. It was my purpose to provide steam power plant designers and operators with a reasonably simple and yet sufficiently accurate means to analyze their installations and thus reduce the incidence of troubles arising from transient conditions.

Simple as these calculations have become, I have found that even further simplification is possible and it is with the description of this simplification that this article will be concerned.

Let us first briefly review the methods in use at present. Following a sudden load reduction, the pressure in the deaerating heater will decay rather rapidly, reducing the pressure at the suction of the boiler feed pump correspondingly. Unfortunately, until the suction piping has been completely voided of the feedwater it contained prior to the load reduction, its temperature and hence vapor pressure will not be reduced. As a consequence, the available NPSH will diminish until such time as it may be quite insufficient to provide adequate pump operation, the pump will flash and serious damage may be incurred.

The analysis of the adequacy of an installation has been based on determining the *allowable* and the *actual* rates of pressure decay in the deaerating heater, of comparing these two rates at different rates of feedwater flow and of determining at what rate of flow the *actual* rate of pressure decay will exceed the *allowable.* Based on a number of simplifying assumptions, formulas were developed for the *allowable* and the *actual* rates of pressure decay. These formulas are reproduced below:

$$\text{Allowable } \frac{dp}{dq} = \frac{-H_x}{Q_s} \qquad (1)$$

where: dp/dq = Rate of pressure decay in ft/gal pumped

H_x = Available excess NPSH in ft = $H_a - H_s$

H_a = Available NPSH in ft at dq/dt gpm = $Z - h_f$

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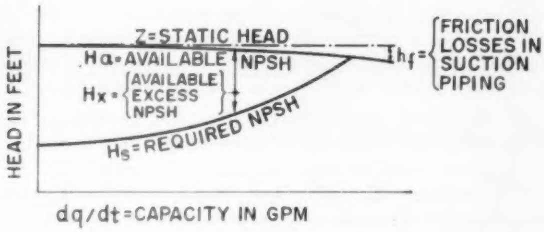

Fig. 1—Determination of H_x, excess available NPSH

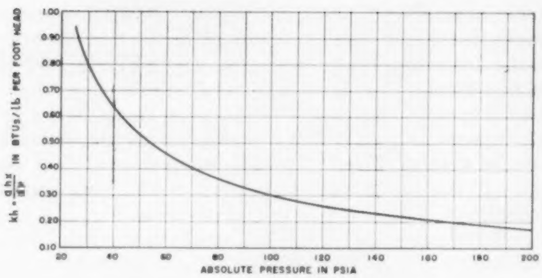

Fig. 2—Enthalpy change with change of pressure for water

Z	=	Static head in feet or vertical distance between water level in heater and pump centerline
h_f	=	Friction losses in suction piping at dq/dt gpm, in ft
H_s	=	Required NPSH in ft at dq/dt gpm
dq/dt	=	Pump flow in gpm
Q_s	=	Volume of the suction pipe, in gal

In turn,

$$\text{Actual } \frac{dp}{dq} = -\frac{(h_{zo} - h_{c2})}{K_h Q_h} \qquad (2)$$

where:

h_{zo}	=	Enthalpy of feedwater in heater prior to load reduction, in Btu/lb
h_{c2}	=	Enthalpy of condensate coming to heater, under final load conditions, in Btu/lb
K_h	=	dh/dp in Btu/lb per foot of absolute pressure, at steam conditions prior to load reduction (see Fig. 2)
Q_h	=	Volume of feedwater heater in heater storage, in gal

A typical plot of *allowable* and *actual* rates of pressure decay in a given installation is presented on Fig. 3. The comparison of the superimposed curves establishes the range of flows which are safe during the transient conditions which follow that particular load reduction. It shows, in this case, that with two pumps running, all flows below 1520 gpm total result in safe operation, while flows above 1520 gpm total would result in an *actual* pressure decay rate which would exceed the *allowable*. While the construction of these two curves of *allowable* and *actual* pressure decay rates is relatively simple, it is also possible to apply the information we have developed to determining a sound method of sizing the deaerator storage volume.

Until the effect of the deaerator heater storage volume on the *actual* pressure decay heater caused by a sudden load reduction was clearly understood, this storage volume was selected strictly on the basis of providing protection and surge volume against interruption of condensate supply. The choice of storage volume was thus based on personal opinions or past practices and generally ranged anywhere from 3 to 15 min storage, depending on the relative degree of optimism or pessimism of the designer.

But if we wish to apply the knowledge developed in predicting the *actual* rate of pressure decay, we can determine with a reasonable degree of accuracy the minimum size of heater storage necessary to avoid suction disturbances caused by a sudden load drop.

There are two separate approaches to this problem. The first approach consists of selecting a desirable size of suction piping and, using equation (1) of determining the value of the *allowable* rate of pressure decay at the expected maximum feedwater flow which may occur after a sudden load drop. This will correspond to the maximum permissible value of the *actual* rate of decay. The required size of deaerator storage can then be solved from the following equation:

$$\textit{Minimum } Q_h = \frac{(h_{zo} - h_{c2})}{K_h\, x \text{ allowable } \dfrac{dp}{dq} \text{ at max. flow}} \qquad (3)$$

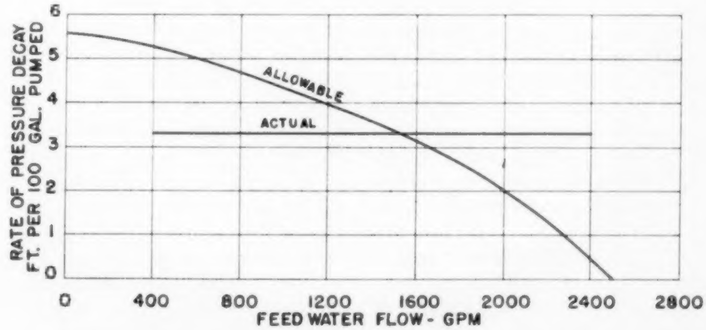

Fig. 3—Comparison of decay rates per unit volume pumped (2 pumps Running)

For example, let us assume that we are dealing with a pump installation such that the *allowable* rate of decay in the deaerating heater is that shown on Fig. 3. Let us further assume that the maximum feedwater flow after a load drop with two pumps running can reach 2000 gpm. The maximum rate of *actual* pressure decay must therefore be held to 2 ft per 100 gal pumped, or 0.02 ft per gal. If the initial heater pressure is 79.2 psia (initial enthalpy = 281.1), the value of K_h from Fig. 2 is 0.36 Btu/lb/ft head. If the final condensate enthalpy is 168 Btu/lb, we can now substitute in equation (3) and

$$Minimum\ Q_h = \frac{(281.1 - 168)}{0.36 \times 0.02} = 15{,}710 \text{ gal}$$

An alternate approach for determining the required size of heater storage is suggested by the fact that the real factor that establishes the adequacy of an installation from the point of view of suction conditions after a load drop, is the ratio between the deaerating heater storage capacity and the suction piping volume. If we equate equations (1) and (2) and transpose, we obtain a relation for the desired ratio between these two volumes:

$$Minimum\ \frac{Q_h}{Q_s} = \frac{(h_{zo} - h_{c2})}{K_h\, H_z} \tag{4}$$

All that it is now necessary to do is to assume the maximum feedwater flow after a sudden load drop, determine the corresponding value of H_z and, substituting the proper values into equation (4), establish the minimum permissible ratio between the volume of the deaerating heater storage (Q_h) and that of the suction piping (Q_s).

In order to appreciate the significance of the relationship expressed in equation (4) we must examine the behavior of each term of the equation with respect to the variants of a steam power plant and the effect of this behavior on the term Q_h/Q_s which we are investigating.

If we accept the thought that the maximum disturbance to the suction conditions occurs in the event of a complete turbine trip-out, then the value of "h_{c2}" corresponds to the condensate temperature as it leaves the condenser hotwell and is therefore fixed within rather narrow boundaries. Assuming that condensate temperatures may vary between 90 F and 115 F, h_{c2} will vary from 58 to 83 Btu/lb. On the other hand, the variation in h_{zo} is greater, since it will be as high as 363 for a heater operating at 220 psia (390 F) and as low as 196 if the heater pressure is 20 psia (228 F).

As a result, the value of minimum Q_h/Q_s increases very rapidly as the pressure at the deaerating heater increases. As a matter of fact, it does so for a second reason, since the value of K_h decreases rapidly with increasing heater pressures, as can be seen from Fig. 2.

The value of the excess available NPSH, "H_z" affects the minimum ratio of heater storage to suction piping volume (Q_h/Q_s) as well as the heater storage volume for a given minimum ratio. This is because in the relationship we have derived, Q_h/Q_s minimum varies inversely with H_z, but if H_z increases for a given installation, this increase automatically increases the volume of the suction piping.

But whatever method is finally used by the steam power plant designer, it is important to remember that one cannot dismiss this problem without risking to endanger the installation every time that a sudden load reduction takes place.

Nuclear Progress Summarized

The atomic industry closed out 1959 and looked ahead to the next decade encouraged by evidence of increasing willingness of reactor manufacturers to offer qualified guarantees of nuclear power costs, by a ten year U. S. Atomic Energy Commission plan for attainment of economic nuclear power and by continuing progress in construction and operation of actual nuclear power projects and programs, according to a year-end summary issued by the Atomic Industrial Forum.

Dampening factors for certain segments of the industry noted by the Forum include a slow-down in the Euratom nuclear construction program, cutbacks in certain U. S. government reactor development programs, over-production of uranium, and a more conservative appraisal of the current value to industry from utilization of radiation and radioisotopes.

In its summary, the Forum noted that several major reactor system suppliers are indicating cost guarantees on nuclear power plant construction factors over which the manufacturer could maintain control. 1959 also saw the U. S. AEC announce details of technical studies upon which it is basing a ten-year government plan for nuclear power development programs to be carried out both at government facilities and at private sites. AEC estimates showed that at least one system, pressurized water, could be developed to the point of being nearly competitive with conventionally-fueled plants by 1965.

Other positive highlights of the past year included: startup of the privately-financed Dresden (Illinois) nuclear power station, the world's largest operating power reactor; launching of the N. S. Savannah, the world's first-nuclear-powered commercial cargo or passenger-carrying vessel; addition of five more atom-powered submarines to the nuclear Navy; and sea trials of the Soviet Union's atom-powered icebreaker, the Lenin.

The Forum review, on the other hand, indicated little encouragement to American hopes for an expanded nuclear equipment market as a result of the Euratom program, with only one of five European reactor construction proposals meeting dead-line requirements of the Euratom invitation, that one being the SENN plant in Italy.

Applications of radiation received a setback with deferment of Army plans to operate an ionizing radiation center for radiation sterilization of food. Relatedly, an AEC-sponsored survey revealed that the total net savings realized by major industries from using radioisotopes amounted to just under $40 million annually. This was approximately one-tenth of AEC's 1957 estimate and is accordingly worthy of note.

Design and Selection of Hyperbolic Cooling Towers*

By R. F. RISH† and T. F. STEEL†

Central Electricity Generating Board

The structural design of hyperbolic cooling towers is discussed with particular reference to wind stresses in the shell. Formulae are given to enable the size of cooling towers to be determined for a given cooling duty, and a method outlined for the selection of the most economic duty.

THE first hyperbolic natural draft reinforced concrete cooling tower was designed by Prof. van Iterson of the Dutch State Mines and installed at the Emma Colliery in 1916. Towers of this type were installed at Lister Drive Power Station in Liverpool in 1925 and since then have become standard practice in British power stations where cooling towers are required. These early towers are still in use, having shown great reliability in service, little maintenance having been required apart from the replacement of decaying timber in the cooling stack. Recent advances in the fields of timber preservation and the use of alternative materials promise to extend the life of the whole of a modern hyperbolic cooling tower to the working life of its associated power station.

The largest tower so far constructed is 340 ft high and 260 ft base diameter and will cool the circulating water for a 200-MW set. Most of this structure is the empty shell but the lower portion contains the cooling stack over which the warm water is distributed by a pipe and nozzle system 32 ft above the ground. The lower portion of the shell is open to allow the air access to the cooling stack, the shell being supported on legs that are inclined to resist the shearing force due to the wind. Beneath the tower a pond is constructed to catch the falling water and return it to the circulating water system. An example of a modern hyperbolic cooling tower is shown in Fig. 1.

The function of the cooling stack is to increase the surface area between the water and the cooling air, either by breaking the water up into droplets or by spreading it over a large area in the form of films. It is important that this be achieved without offering too great a resistance to the movement of air.

Fig. 1—Typical modern natural draft cooling tower

As the warm water falls through the stack it gives up its heat to the air. The air leaving the stack inside the shell is lighter than the ambient air and a draft is created by chimney effect. This mechanism differs from that of a mechanical draft tower in that the cooling is dependent on the dry bulb temperature as well as the wet bulb temperature, the draft for a given wet bulb temperature increasing with decrease of dry bulb temperature (i.e., being better in humid conditions). Above the distribution pipes a spray eliminator screen is constructed to catch the fine droplets of water which would otherwise be picked up by the rising air-stream and deposited over the neighborhood. As the plume of vapor is discharged at a great height there is no nuisance due to ground fog, nor is there any risk of the performance of the tower being affected by the recirculation of the air leaving the top of the tower.

The cooling characteristics of a hyperbolic type natural draught cooling tower are different from those of a mechanical draught installation. If it is desired to compare the economics of the two types of tower it is neces-

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sary first to select the most economic cooling water arrangement for each type of tower by a method similar to that outlined in the paper. The most economical arrangements of both systems can then be compared taking into account such factors as overall capital cost, condenser vacuum, auxiliary power required, reliability, cost of maintenance and ground area. It can be quite misleading to compare the economics of the two types of cooling tower on the basis of a common cooling system and tower duty.

Design of Shell by Membrane Theory

There is no thermodynamic reason why the shell of a natural draught cooling tower should not be cylindrical. This shape simplifies both the design and construction of the shell and cooling towers have been built in Germany on these lines.

However, the momentum of the air entering the shell carries it into the center to form a *vena contracta*[1] whose diameter depends on the ratio of tower diameter to height of air inlet. A considerable saving in shell surface area and volume of concrete can then be made by tapering the shell in to the diameter of the vena contracta. This stiffens the shell against wind forces and opening out the shell above the throat stiffens the shell even more.

The analysis of the effect of wind forces on a shell can be carried out by the "membrane" theory, which assumes that the thickness of the shell is so small compared with its diameter that the wind forces are resisted only by direct tensions, compressions and shearing forces in the direction of the shell itself. The analysis of the true hyperbola of revolution is complex and is usually carried out using a step-by-step method. A very good approximation however can be made by assuming the shells to be made up of two truncated cones with a cylinder in between. Towers of this "diabolo" shape have been built in the past but as they are rather ugly and show no other advantage than ease of calculation they are no longer being constructed. The simple case of the cylinder is dealt with to demonstrate the method, and the analysis of a tower built up of two truncated cones is shown in Appendix 1.

[1] Editor's Note: An hydraulic term for any of the contracted parts of minimum size of a jet of Fluid discharging from an orifice or an aperture;—usually restricted to the one nearest the orifice.

Analysis of Cylindrical Shell

Consider the small element of shell shown in Fig. 2 acted upon by a normal wind pressure p, which is resisted by vertical loads/ft, V, and $(V + dV)$, horizontal ring loads/ft, t and $(t + dt)$ and shearing loads/ft, s, and $(s + ds)$.

These forces are in equilibrium.

Resolve vertically

$$V.Rd\beta + S\ dh = (V + dV)\ R\ d\beta + (S + ds)dh$$

Whence

$$ds/d\beta + R\,dV/dh = 0 \qquad (1)$$

Resolve horizontally

$$t\ dh + S.Rd\beta = (t + dt)dh + (S + ds)\ R\ d\beta$$

Whence

$$dt/d\beta + R\ ds/dh = 0 \qquad (2)$$

Resolve perpendicularly to element

$$t = -PR \qquad (3)$$

Solving these equations will give values of v, s and t at any depth h from the top of the shell and any angle β to the wind. From (3) $dt/d\beta = -R\,dP/d\beta$). Equation (2) becomes

$$-\ R\ dP/d\beta + R\ ds/dh = 0$$
$$(ds/dh = dP/d\beta)$$
$$(S = h\ dP/d\beta + \text{const.})$$

At top of shell $S = 0$ (i.e., when $h = 0$, $S = 0$) $\therefore$ const. $= 0$

$$\therefore\ S = h\ dP/d\beta$$
$$ds/d\beta = h\ d^2P/d\beta^2 \qquad (4)$$

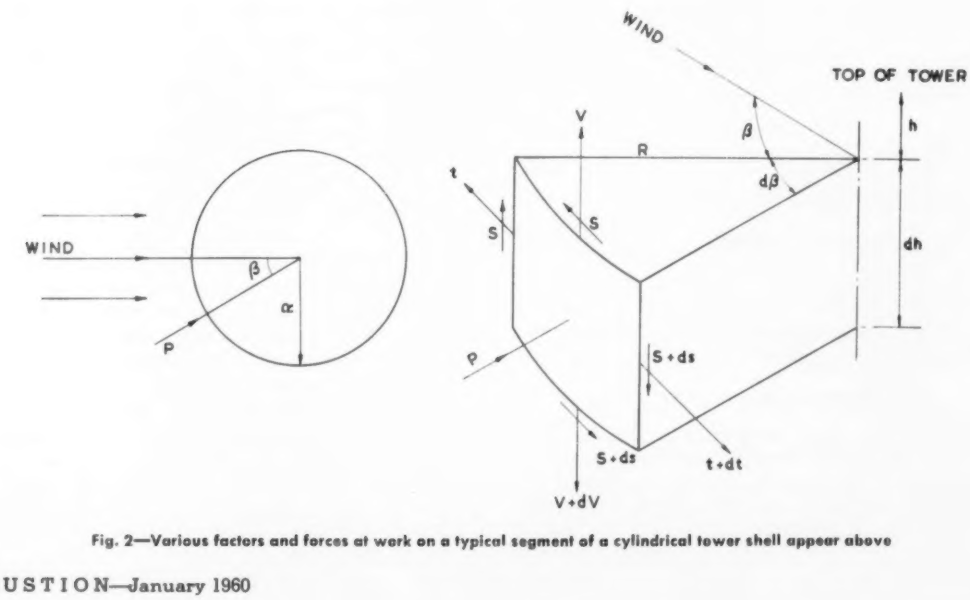


Fig. 2—Various factors and forces at work on a typical segment of a cylindrical tower shell appear above

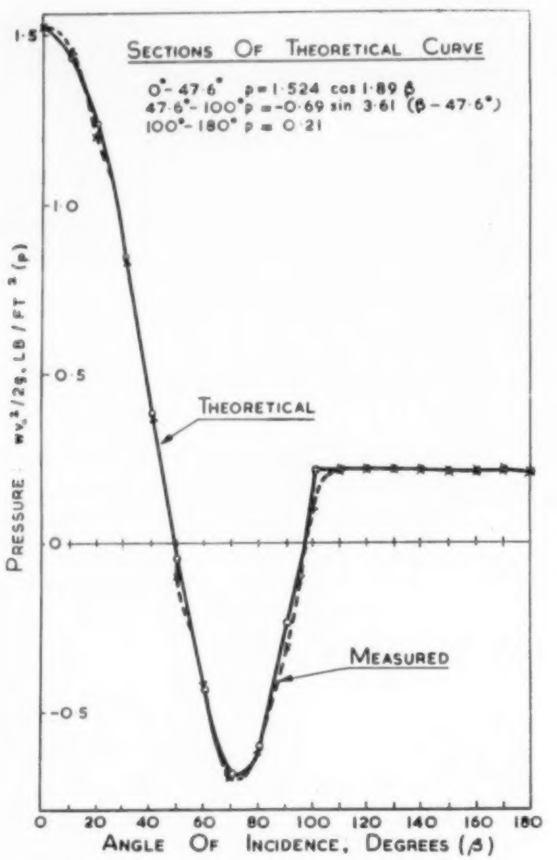


Fig. 3—Distribution of wind force obtained from a 26½ in. model sheet metal cooling tower under high pressure, i.e. increased density and high Reynolds Numbers

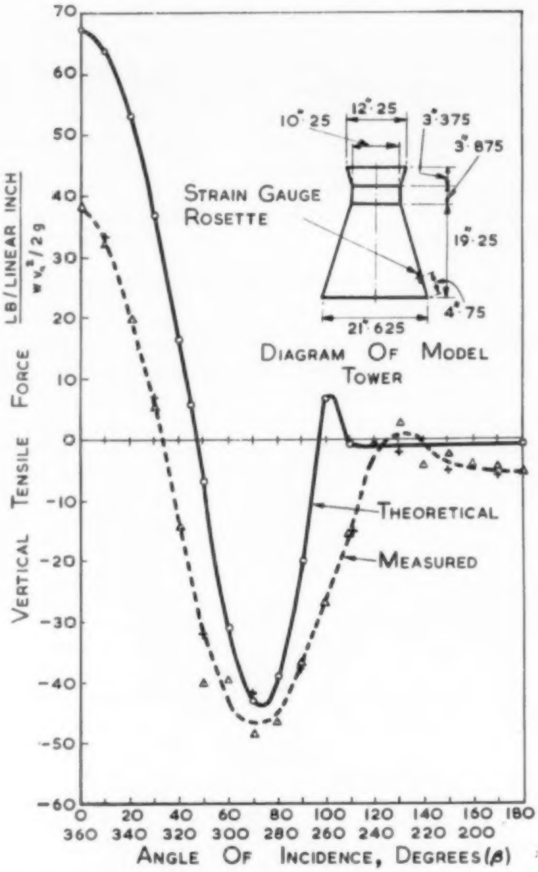


Fig. 4—To check the validity of the authors' theory on the membrane approach as a means of determining wind effects the test data of Figs. 4, 5 and 6 were obtained

Equation (1) becomes $h\, d^2P/d\beta^2 + R\, dV/dh = 0$ or

$$dV/dh = -h/R \times d^2P/d\beta^2$$
$$V = -h^2/2R \times d^2P/d\beta^2 + \text{const.}$$

At top $V = 0$ (i.e., when $h = 0$, $V = 0$) $\therefore$ const. $= 0$

$$\therefore\ V = -h^2/2R \times d^2P/d\beta^2 \qquad (5)$$

The three equations are then

$$t = -P\,R \qquad (3)$$

$$S = P'h \qquad (4)$$

where $P' = dP/d\beta$.

$$V = -P''\,h^2/2R \qquad (5)$$

where $P'' = d^2P/d\beta^2$.

In order to solve the above equations it is necessary to know how the wind pressures are distributed around the tower shell, i.e., how P varies with β.

Wind Pressure Measurement

Although there have been some experiments on cylinders and model cooling towers to find out the pressure distribution, in no case did the Reynolds Number of the test approach that attained at high winds on a modern hyperbolic cooling tower, the difficulty being that with a model scale of $^1/_{100}$ the wind velocity would have to be 7000 mph to correspond with a 70 mph wind at the same pressure. It is known that the vibration of a chimney can extend subcritical distributions of pressure to Reynolds Numbers greater than that which would have been expected and for this reason it was desired to obtain readings at the highest Reynolds Number attainable. A model cooling tower $26^1/_2$ in. high was built out of sheet metal with pressure measuring points both inside and outside the shell, and tested under pressure (i.e., at increased density and Reynolds Number) in the variable-density tunnel belonging to the Aerodynamics Division of the National Physical Laboratory at Teddington, England.

The distribution of wind force obtained from these tests is shown in Fig. 3. Although it would have been more accurate to use one equation to cover the whole curve, in view of the successive differentiating required it was felt that it would be easier and sufficiently accurate to split up the curve into a number of parts each covered by a simple expression, and this has been done.

For example, a cylindrical concrete cooling tower shell is 189 ft 4 in. diameter 219 ft 6 in. from the top of the ring beam to the top of the tower and 9 in. thick. What are the maximum stresses produced by a uniform wind speed of 90 mph?

$$t = -PR = -94.66P$$
$$S = P'h = +219.5P'$$
$$V = -P'' h^2/2R = -255\ P''$$

The maximum values of t and v will occur at $\beta = 0$ and the maximum value of S at $\beta = 47.6$ deg.

The wind pressure on a vertical plane surface at right angles to the wind will be $wv^2/2g$ where w is the density of the air and v the velocity of the wind. With a velocity of 90 mph a temperature of 59 F and a barometric pressure of 29.53 in. of mercury, this will equal 20.75 lb/,sq ft

$$P = 20.75 \times 1.524 \cos 1.89\beta = 31.6 \cos 1.89\beta$$
$$P' = -31.6 \times 1.89 \sin 1.89\beta = -59.7 \sin 1.89\beta$$
$$P'' = -59.7 \times 1.89 \cos 1.89\beta = -113 \cos 1.89\beta$$

$\beta = 0$ deg $\quad P_0 = 31.6 \quad P_0' = 0 \quad P_0'' = -113$
$\beta = 47.6$ deg $\quad P_{47.6} = 0 \quad P'_{47.6} = 59.7 \quad P''_{47.6} = 0$

t at $\beta = 0$ deg $= 2990$ lb/ft (compression)
S at $\beta = 47.6$ deg $= 13{,}100$ lb/ft
V at $\beta = 0$ deg $= 28{,}800$ lb/ft (tension)

A 9 in. shell weighs 112.5 lb/sq ft hence

V dead load $= 219.5$ ft $\times$ 112.5 lb/sq ft $= 24{,}700$ lb/ft

The maximum resultant vertical tension is 4100 lb/ft.

If the permissible tensile stress in the steel is taken as 24,000 lb/sq in. the area of vertical steel required per foot of the shell is 0.17 sq in. or $^3/_8$ in. bars at 8 in. c/c.

It is interesting to compare the above results with those obtained for a tower of similar height and radius at the section considered, but made up of two truncated cones with a cylinder in between as shown in Fig. 4. The equations are:

$$t = -\ 98.9\ P$$
$$S = +\ 121.4\ P' = 5.95\ P''' + 0.214\ P^{V}$$
$$V = -28.5\ P\ -190.2\ P''\ -18.9\ P^{IV}\ -\ 0.63\ P^{VI}$$

which gives the loads per foot with wind pressures as in the previous example.

t at $\beta = 0$ deg $= -3125$ lb/ft (compression)
S at $\beta = 47.6$ deg $= -6143$ lb/ft
V at $\beta = 0$ deg $= 13470$ lb/ft (tension)

It will be seen that the shear and vertical stresses are reduced by over 50 per cent due to the "hyperbolic" shape of the shell compared with a cylinder of the same height and base diameter.

Strain Gage Measurements

In order to check the validity of the theory, strain gages were fixed to the shell of the model tower and readings taken under load in the wind tunnel. Owing to the difficulties of temperature compensation in the tunnel the accuracy of the readings was not very great, nevertheless it is considered that the results support the membrane theory as being a reasonable method for the analysis of cooling tower shells. The results obtained from one set of gages which may be regarded as typical are shown in Figs. 4, 5 and 6 compared with the theoretical results.

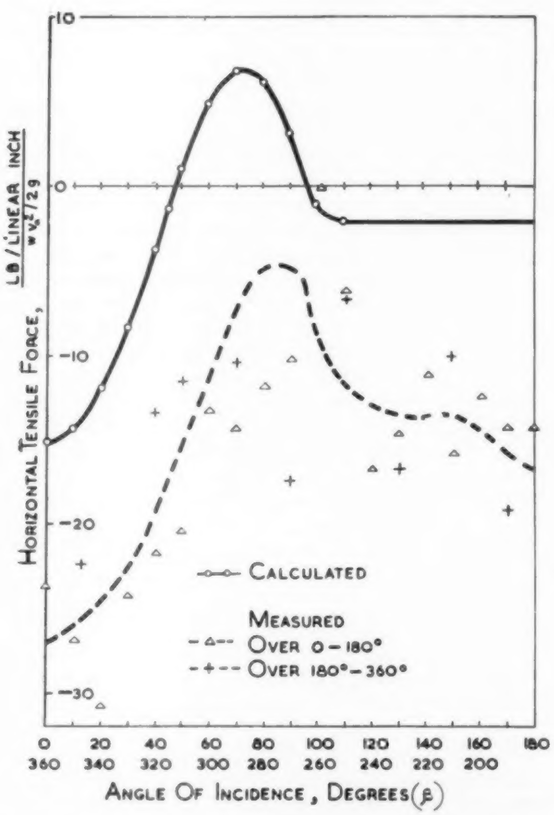


Fig. 5—Comparison of the above results compares calculated values of horizontal tensile forces against those measured

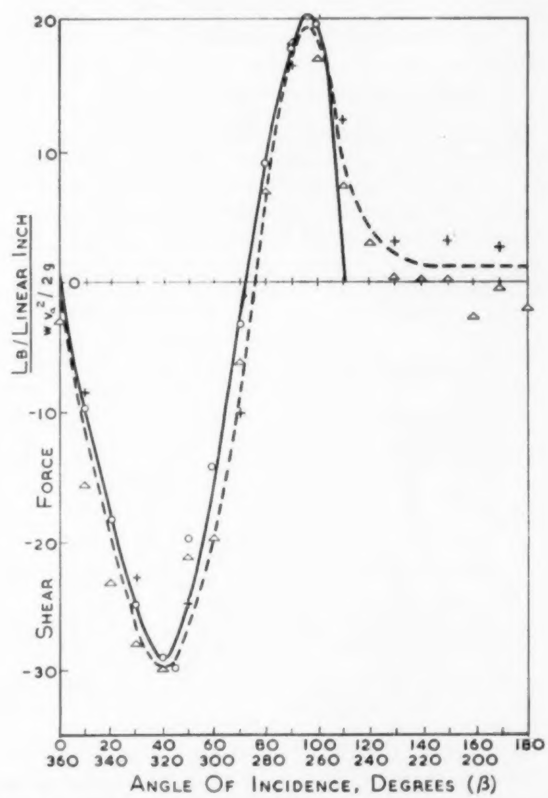


Fig. 6—The data from each of the Figs. 4 and 5 together with the above represent typical values obtained by strain guages fixed to the shell of the model tower

Moisture Movement Stresses in Shell

In operation the tower shell becomes saturated with water which causes it to swell. According to Murdock (1)[2] a typical 1:2:4 concrete with a w/c ratio 0.6 will expand 0.028 per cent. If the outside of the tower shell is dried by a hot sun or by wind while the inside remains saturated, assuming a linear distribution of strain, the maximum tensile and compressive stresses in the section will be $\frac{1}{2} \times 0.0028\, E$ where E is the Young's modulus of the concrete. If E is taken as 2.5×10^6 lb/sq in. the tensile stress produced is 350 lb/sq in. which may be sufficient to crack the concrete.

There are two possible approaches to this problem. The provision of 0.3 per cent horizontal reinforcement on the outside of the shell with 1 in. cover will reduce the moisture movement stresses to acceptable limits. This is fairly expensive as it involves a considerable increase in the reinforcement of the shell. Also it involves placing long heavy bars on the outside of the vertical reinforcement, whereas the working platform is usually on the inside.

The alternative is to paint the inside of the shell with bituminous paint to prevent the moisture reaching the concrete. As the scaffolding used in the construction of the shell is moving upwards fairly rapidly, to avoid heavy additional scaffolding costs the paint has to be applied while the concrete is still green. Not many paints will stand up to these conditions and great care must be taken in the selection and application of a suitable paint. Of a large number tested by Parkinson (2) only two proprietary brands were found to be satisfactory and these are now specified where painting of cooling towers is required.

Design of Ring Beam and Tower Legs

The ring beam is the intermediate portion between the tower legs and the main portion of the shell. The thickness of this ring beam is sometimes gradually tapered from the bottom, where it has to be thick enough to accommodate the tower legs, to the $4\frac{1}{2}$ in. or 5 in. of the

[2] Numbers in parentheses refer to similar numbered entries in References at the close of the article.

Fig. 8—Precast concrete cooling stack

shell. In other designs the ring beam is made of constant thickness with an abrupt transition to the shell. The stresses set up in the ring beam may be calculated from deep beam theory. The load on the inclined legs of the tower is made up of the dead load of the shell and ring beam together with the shear and vertical wind loads obtained from the membrane theory. The tower legs should preferably be in the same line as the shell. Sometimes however, where space is limited, the inclination of the legs is changed to produce a smaller pond as in Fig. 7, Tower B. In this case tension is set up at the change of direction and although it is possible to insert sufficient steel at this point to take the stress there is some danger of cracking taking place due to tensile stresses in the concrete. This danger can be overcome by changing direction gradually instead of abruptly, the length of the curve being such that the tensile strength of the concrete is not exceeded. This is shown in Fig. 7, Tower C.

The load at the base of the tower legs can be taken on the pond wall acting as a lower ring beam or on independent footings.

The spray from the circulating water tends to evaporate from the tower legs leaving concentrations of salts which sometimes cause deterioration of the concrete.

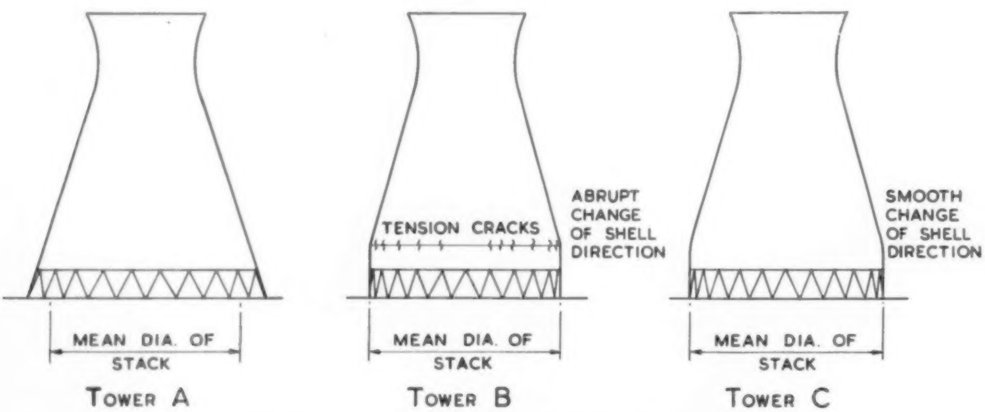


Fig. 7—The various approaches to design of tower legs and ring beam show above. A carries out the tower's normal contour to ground. B attempts to fix the ring beam at a constant thickness and indicates cracks from the abrupt change of shell direction. C attempts the same transition but effects it through a gradual change in shell direction

To allow for attack of this kind the legs should be designed with adequate cover, the strength of the concrete in this cover being neglected in the stress calculations. If there is much sulfate in the circulating water it is advisable to paint the legs with bituminous paint.

Cooling Stack

Before the war timber was universally used as the supporting structure for the cooling stack. Due to decay at the joints, however, the timber stacks gradually became unstable and the structures have usually had to be replaced once or twice during their working life of 30 years. Owing to the shortage of suitable timber during the war precast concrete supporting structures were brought into use; it was then found that the extra capital cost was more than compensated for by the longer life. An example of a precast concrete supporting structure is shown in Fig. 8.

Timber (*Pinus Silvestris*) is usually used for the splash bars or lathing and a good deal of research has been carried out by the Generating Board and the timber treatment firms into the cause and prevention of decay. Suitable water borne preservatives which combine with the timber to form insoluble salts have given results which promise a considerable extension of life to cooling tower timber when used with the necessarily high degree of impregnation.

The other approach to the problem of packing decay is the use of a packing which does not deteriorate. Glass lathing has been investigated and a highly efficient tower using this material could no doubt be constructed. However, there is an understandable reluctance to use glass on a big scale because of the danger of breakage and falling lathing.

Asbestos-cement sheets as a packing material have also been studied and appear promising. An experimental packing of one layer of corrugated sheets was installed in a small cooling tower in 1945, again stimulated by a shortage of timber. As insufficient depth had been provided this tower gave a disappointing performance but the packing has stood up to cooling tower conditions very well, no sign of deterioration having occurred in thirteen years of continual use. Recent analysis has shown that with a double layer of corrugated asbestos sheets a most efficient tower should be obtained. A tower to this design is being installed in association with a 120-MW set at the present time and should be ready for testing soon. Although the asbestos-cement cooling stack was expensive compared with a timber stack for the same duty, the smaller size of tower required brought the total first cost below that of the cooling tower design with the conventional timber stack. Asbestos-cement packed towers using one layer of flat sheets have also been built in Germany to the design of Dr. W. Otte.

The asbestos-cement packed tower (together with some modern developments in timber packing) has its stack enclosed entirely in the shell, the tower beneath the air opening being empty except for the columns supporting the stack. This has the advantage that the cooling takes place in counter-flow and enables the thermal design theory to be employed with greater accuracy than when some cross-flow takes place. This does not imply that the performance of mixed flow towers cannot be assessed, as a great deal of empirical knowledge has been gained from past experience, but it is simpler to predict the effect of innovations in a counter-flow tower. All counter-flow towers depend for their efficiency on a low overall resistance to air flow and it is important that an adequate air-opening should be employed.

Icing

In cold weather the droplets in the outer periphery of the tower tend to freeze, icicles build up on the packing and sheets of ice form which can damage the packing and obstruct the air flow. In most towers anti-freeze systems are now installed so that hot water can be drawn from the inlet pipe and sprayed over the periphery. A considerable quantity of water is required in really cold weather (probably about 25 per cent of the circulation) and in some cases the capacity of the anti-freeze system has proved insufficient. Additional water can usually be obtained in a center feed tower by opening the ends of the distribution pipes whereupon the water will run down the tower wall on to the periphery of the stack.

If a counter-flow packing is employed there is no hold for the ice to build up on and serious icing is unlikely to occur.

Makeup and Purge

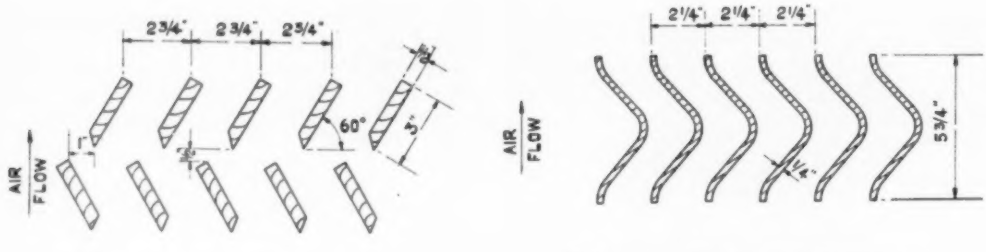
During the operation of a cooling tower a small proportion (about 1 per cent) of the circulating water is continually being evaporated and has to be replaced. The salts in the water are left in solution and if the water is not purged concentrations will be reached at which precipitation will occur in the form of scale. Additional makeup water is therefore required above the evaporation loss, the surplus usually spilling over the purge weir to waste. If the purge quantity is twice the evaporation loss the concentration of salts cannot increase to more than 50 per cent above that of the makeup water. Chlorination is also required in certain circumstances to keep algae from growing on the splash bars.

Where an installation rejects its heat solely to cooling towers and the makeup water contains silt which may cause abrasion to condenser tubes, the cooling towers ponds form a useful settlement area. The ponds are therefore designed so that a reasonable quantity of silt can be collected and removed.

Where a limited cooling source is supplemented by towers and the water is silty, saucer-shaped ponds are employed with outlet troughs running across the diameters. Water falling from the cooling stacks will then wash any silt into the outlet troughs and keep the settlement in the ponds to a minimum.

Spray Eliminators

The finer droplets of water produced by the nozzles and the cooling stack are picked up by the rising airstream and can create a public nuisance if not intercepted. The problem was not tackled effectively until the nationalization of the British electricity supply industry in 1948 when a large number of eliminator screens was examined in a test tower and the most suitable selected as a standard eliminator screen to be employed in power station cooling towers. This double layer louvred screen shown in Fig. 9(*A*) is low in first cost, effective in practice and has the low pressure drop of only three velocity heads. As the column of air inside the tower is lightened



(a) Louvre Eliminator (b) Asbestos Cement Eliminator

Fig. 9—Spray eliminators were never too effective until 9A was developed in 1948. The asbestos cement eliminators, Fig. 9B is expected to prove more effective with a lower pressure drop and greater durability

by the elimination of droplets no deterioration in performance is produced by the introduction of an eliminator screen of this type. An account of this work has been given by Chilton (3).

A new development has been the asbestos-cement eliminator screen shown in Fig. 9(B) for which a British patent is pending. This is more effective, has a lower pressure drop, and is more durable than the timber screen, but is rather more expensive in first cost.

Performance of Cooling Towers

Chilton (4) showed that the duty coefficient D of a tower is approximately constant over its normal range of operation and is related to the tower size by an efficiency factor known as the performance coefficient C as follows:

$$D = \frac{\Delta\sqrt{H}}{C\sqrt{C}} \tag{6}$$

where A is the base area of the tower measured at pond sill level and H is the height of the tower measured above sill level.

The duty coefficient may be worked out from the formula

$$W_{L/D} = 90.59\,\frac{\Delta h}{\Delta T}\sqrt{\Delta t + 0.3124\,\Delta h} \tag{7}$$

where Δh is the change in total heat of the air passing through the tower, ΔT is the change of temperature of the water passing through the tower and W_L is the water load in lb/hr. The air leaving the packing inside the

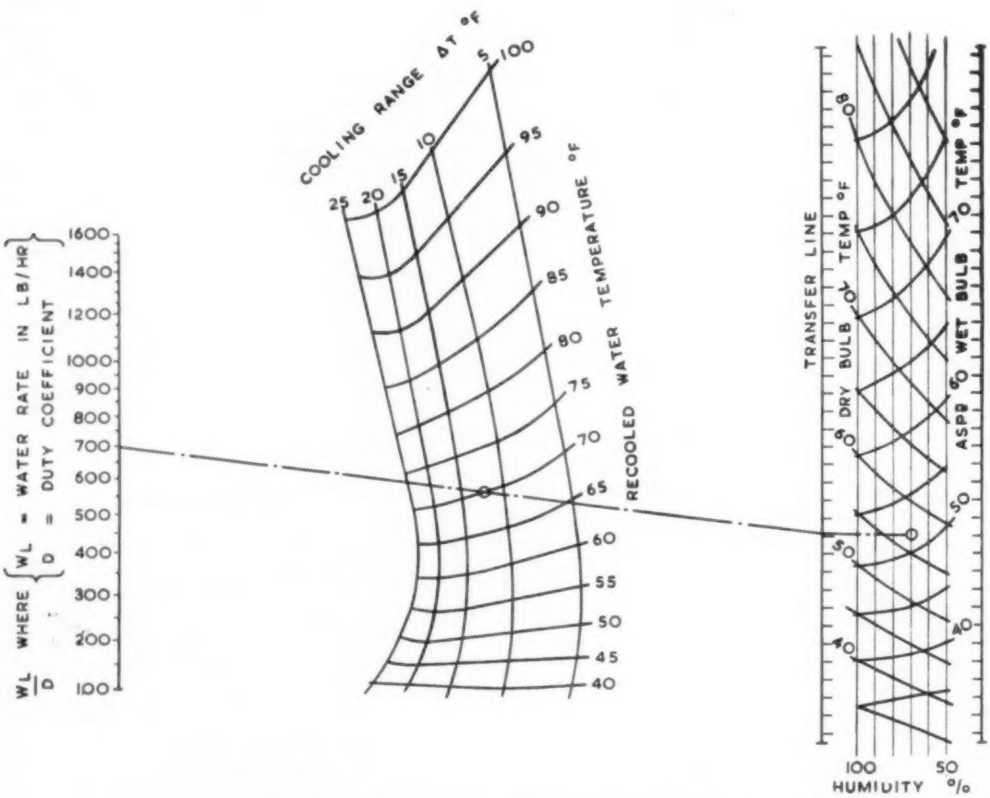


Fig. 10—Universal performance chart for natural draft cooling towers

tower is assumed to be saturated at a temperature half way between the inlet and outlet water temperatures. A divergence between theory and practice of a few degrees in this latter assumption does not significantly affect the result as the draught component depends on the ratio of the change of density to change of total heat and not on the change of temperature alone. As an example:

Temperature of water to tower	= 82 F
Recooled water temperature	= 70 F
Temperature range ΔT	= 12 deg F
Dry bulb air temperature t_2	= 57 F
Aspirated wet bulb air temperature t_2'	= 51.7 F
Water loading to Tower W_L	= 38.2 million lb/hr

$$
\begin{array}{llll}
t_1 = (82\text{ F} + 70\text{ F})/2 = & 76\text{ F} & h_1 = & 32.1 \text{ from hygro-} \\
t_2 = & \underline{57\text{ F}} & h_2 = & \underline{13.6} \left\{ \text{metric} \right. \\
\Delta t = & 19^\circ\text{F} & \Delta h = & 18.5 \left. \right\} \text{tables} \\
0.3124\ \Delta h = & \underline{5.8} & & \\
 & 24.8 & &
\end{array}
$$

$$W_{L/D} = 90.59\ \frac{\Delta h}{\Delta t}\ \sqrt{\Delta t + 0.3124\ \Delta h}$$

$$= 90.59 \times \frac{18.5}{12}\ \sqrt{24.8} = 696$$

$$D \quad = 38{,}200{,}000 \div 696 = 55{,}000$$

The performance coefficients usually attained in the past have been in the region of 5.2 where water loadings are over 750 lb/hr/sq ft though new types of packing are bringing this down, i.e., improving it.

Taking a C value of 5.0 and a tower height of 320 ft the base area of the tower will be $55{,}000 \times 5\sqrt{5} \div \sqrt{320} = 34{,}600$ sq ft. This means that the internal base diameter at sill level will be 210 ft. A ratio of height to base diameter of 3:2 is normally employed.

In order to find how a tower of any given duty coefficient will perform under varying conditions of air temperatures, water loadings and temperature ranges equation (7) has been plotted as a nomogram in Fig. 10.

A more detailed version of this nomogram or performance chart may be modified for a particular value of duty coefficient and included in the specification for a cooling tower. After the tower is constructed it can then be tested within the range of conditions shown on the chart and its success judged from the relation between the recooled temperature attained and that shown on the nomogram for the air temperatures, water loading and temperature range imposed on the tower.

The CEGB's present practice requires the guarantee to cover water loadings between 90 per cent and 110 per cent of the normal water rate, cooling ranges between 2 deg F below and 2 deg F above the normal cooling range, atmospheric wet bulb temperatures between 40 F and 60 F and humidities between 50 per cent and 100 per cent.

It is standard practice to measure the air temperatures at 4 ft above the ground with an aspirated psychrometer and on this basis towers appear to vary considerably in their performance from one test to the next. As the air entering the tower actually comes from a higher level than this and the draft is affected by the air density up to the top of the tower it is evident that more consistent results may be obtained by taking air temperatures at a higher level. The results of two tests taken on one tower are shown in Table 1. More work is continuing on these lines but any rapid change in test procedure is undesirable as most of the test data available have been based on the old method, and the ambient temperature records have also been taken at a standard height of 4 ft above the ground. Owing to the variation in measured tower duty it is advisable for the tower performance to be assessed on the average duty attained on a number of tests spread over the range of the guarantee.

TABLE I—COMPARISON OF TEST RESULTS AT DIFFERENT PSYCHROMETER HEIGHTS.

TEST Nº	WEATHER	TIME	WL LB/HR	ΔT °F	RECOOLED TEMP °F	PSYCHROMETER HEIGHT FEET	WET BULB TEMP °F	DRY BULB TEMP °F	DUTY COEFFIC'T D
1	LIGHT S.E. WIND SKY OVERCAST WITH CLEAR INTERVALS	2 - 3 (AFTERNOON)	45×10^6	16·8	68·8	4	48·8	53·0	70,900
						20	47·2	50·9	65,500
2	DITTO	1 - 3 (NIGHT)	44×10^6	15·9	68·5	4	45·3	47·1	57,300
						20	44·9	46·9	56,500

Wind does not appear to affect the performance of towers significantly, but tests are specified to take place when the wind velocity is less than 15 mph. Towers are tested for not less than four hours with steady operating conditions and the test period is taken as the hour in in which conditions are most steady. If the air temperatures are rising or falling the hold up of water in the pond can produce serious errors in the result. Readings are taken every five minutes and all instruments have to have been calibrated not more than six months before the test. Water flows are usually taken by pitot tube traverses of the conduits, but venturis, weirs and heat balances across the condenser have also been employed.

Spacing of Towers

Figs. 11 and 12 show cooling tower installations in modern power stations, the spacing employed and the relation to the rest of the plant. Towers are normally spaced with one and a half internal base diameters between their centers at which spacing they do not appear to influence each other significantly, but further work into this problem is being planned.

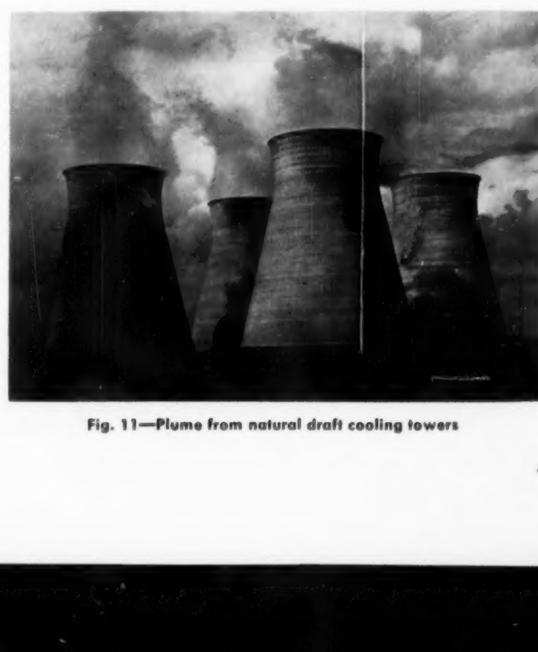


Fig. 11—Plume from natural draft cooling towers

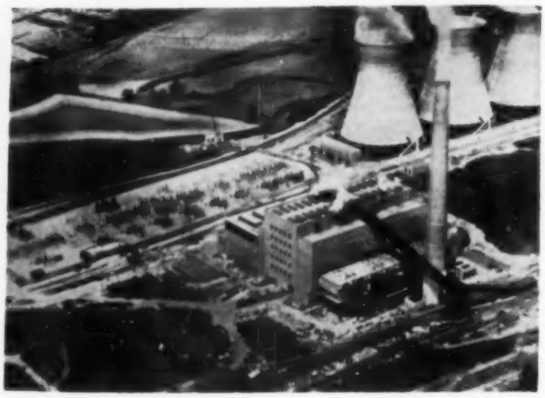


Fig. 12—Cooling tower installation in a modern power plant

Selection of Cooling Water Duty

The selection of cooling tower duty cannot be treated in isolation from the rest of the cooling system. Any change in cooling water quantity will be reflected in the temperature range through which the cooling towers have to recool the water and these variations will have repercussions on the capital cost of the other circulating water plant items, namely, condensing plant, cooling water pumps, pipework and valves and also on the cost of pumping the cooling water. This means that the investigation should not only establish the appropriate cooling tower duty, but also the economic water quantity and temperature levels for the complete condensing system.

The factors to be taken into account in assessing the most economic tower duty are (a) atmospheric temperatures, (b) load factor, (c) turbine exhaust characteristic.

Ambient Atmospheric Temperatures

In Britain the atmospheric temperatures do not differ appreciably over the general areas where adequate fuel supplies (and loading) are relatively close at hand, but where cooling facilities may well be limited. The atmospheric temperatures can, therefore, be averaged without seriously prejudicing the result of such investigation. To take an example, for medium load, i.e., 2-shift operation cooling tower installations, the dry and wet bulb temperatures based on a daily period from 6 a.m. to 9 p.m. are significant, and Fig. 13 shows the monthly average of the returns from 13 meteorological stations over a period of 50 years. When considering an installation remote from these general areas of development reference can be made to the local meteorological records.

Load Factor

The plant load factor may also vary and should be given consideration. In Britain the problem is simplified in that machines are expected to be block loaded which means that partial load conditions can be ignored. The average load factor can usually be predicted on an annual basis, but this should, if a more accurate solution is to be obtained, be broken down to a monthly basis. The shape of the load curve is, of course, significant because the most economic design, even for a given annual load factor, will depend on whether generation is concentrated in one period or spread out over the year. Fig. 14 shows the monthly average load factor for a medium load installation.

Turbine Exhaust Characteristic

The turbine exhaust characteristic establishes the performance level for any given vacuum. Fig. 15 shows the vacuum curves at full load for two 120-MW turbine

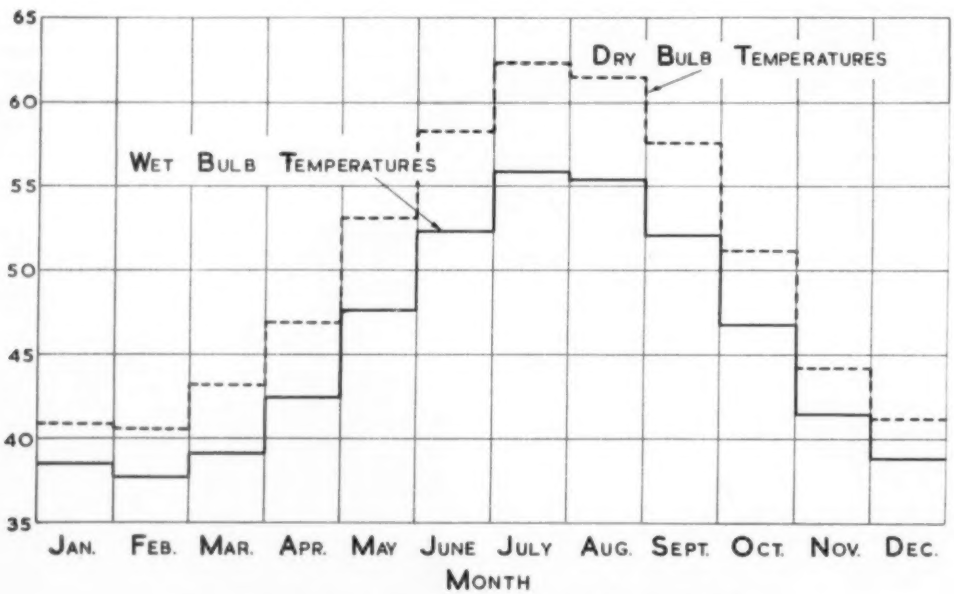


Fig. 13—This 50-year average of 13 meteorological station readings of dry bulb and wet bulb temperatures apply for most of Britain.

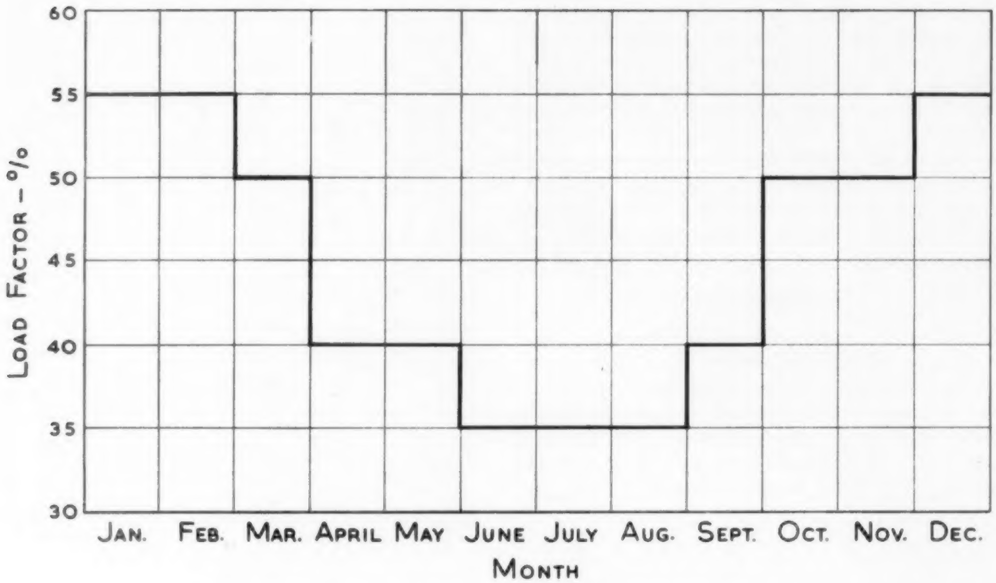


Fig. 14—Monthly average load factor for a medium load installation is pictured above for Britain

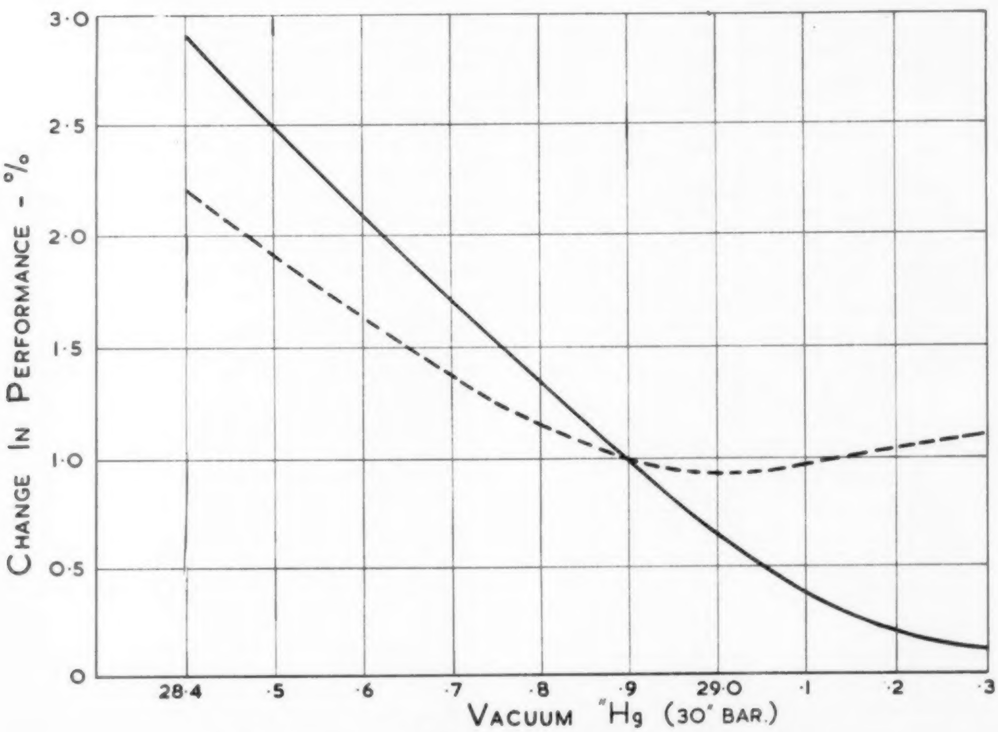


Fig. 15—Vacuum correction curves at full load for two 120 Mw units shows definite variations. Dotted line represents a smaller exhaust

exhaust designs, the dotted line being for the one with the smaller exhaust area.

The average rate of loss in performance for vacua below 28.9 in. Hg is 2.4 per cent per inch Hg for the small exhaust, but there is very little improvement in performance with higher vacua. The large exhaust on the other hand, has an average rate of loss in performance for vacua below 28.9 in. Hg of 3.4 per cent per inch, but there is considerable improvement in performance with higher vacua. It will be noted that it can be quite misleading to assume a constant rate of change in performance with variation in vacuum in order to simply the economic assessment and furthermore the particular turbine exhaust design has a direct bearing on the economics of the cooling system.

Procedure

The method of establishing the economic cooling tower size can perhaps best be illustrated by giving an example of the procedure based on a 120-MW machine.

The monthly averaged atmospheric temperatures and load factors were established and three cooling water quantities namely, 470,000, 587,000 and 783,000 lb per min assumed. The total heat rejection from the machine was known and the cooling water temperature range ($T_2 - T_1$) could be established for the various water quantities, T_1 and T_2 being the condenser inlet and outlet cooling water temperatures respectively.

TABLE II—TOWER NO. 2, FIG. 10, COOLING TOWER RECOOLED TEMPERATURES.

	MONTH	DRY BULB TEMP °F	WET BULB TEMP °F	TOWER No 2 RE-COOLED TEMP °F
$T_3 - T_2 = 6.0$°F	$T_3 - T_1 = 22.0$ °F WATER QUANTITY 587,000 POUNDS PER MIN. $T_2 - T_1 = 16.0$°F			
	JAN.	40.1	38.0	60.4
	FEB	39.8	37.1	60.1
	MARCH	43.0	38.8	61.1
	APRIL	46.6	41.9	63.0
	MAY	53.0	47.4	66.7
	JUNE	59.2	52.9	70.5
	JULY	63.0	56.2	72.9
	AUGUST	61.7	55.5	72.2
	SEPT	57.7	52.1	69.9
	OCT	51.0	46.8	65.9
	NOV.	43.3	40.9	62.2
	DEC	40.1	38.2	60.5
	$T_3 - T_1 = 26.0$°F WATER QUANTITY 470,000 POUNDS PER MIN. $T_2 - T_1 = 20.0$°F			
	JAN	40.1	38.0	58.2
	FEB.	39.8	37.1	58.0
	MARCH	43.0	38.8	58.9
	APRIL	46.6	41.9	60.9
	MAY	53.0	47.4	64.6
	JUNE	59.2	52.9	68.4
	JULY	63.0	56.2	70.8
	AUGUST	61.7	55.5	70.0
	SEPT.	57.7	52.1	67.7
	OCT.	51.0	46.8	63.9
	NOV.	43.3	40.9	60.0
	DEC.	40.1	38.2	58.4

TABLE III—DIFFERENTIAL COSTS IN POUNDS OF COOLING WATER SYSTEMS.

TOWER SIZE	COND^R TERMINAL DIFF $T_3 - T_2$ °F	WATER QUANTITY 1000 LB PER MINUTE	TOWER COST + FIXED PUMPING COST - £	VARIABLE PUMPING COST - £	PIPING, CULVERTS, PUMPS & VALVES - £	CONDENSING PLANT - £	TURBINE RUNNING COST - £	DIFFERENTIAL COST - £
1	6	783	328,000	11,460	21,145	15,700	0	376,305
1	6	587	287,700	9,335	8,880	14,200	32,150	352,265
1	6	470	265,585	3,740	0	0	80,750	350,075
2	6	783	254,000	8,350	21,145	15,700	76,050	375,245
2	6	587	203,700	7,000	8,880	14,200	95,850	329,630
2	6	470	181,585	1,870	0	0	150,150	333,605
3	6	783	216,000	5,240	21,145	15,700	140,350	398,435
3	6	587	175,700	4,665	8,880	14,200	191,850	395,295
3	6	470	153,585	0	0	0	267,350	420,935

Three sizes of cooling tower having duty coefficients of 72,400 (Tower 1), 58,250 (Tower 2) and 45,900 (Tower 3) were considered in relation to the monthly averaged atmospheric temperatures, cooling water quantities and temperature ranges, in order to establish by use of the performance chart Fig. 10, the averaged monthly recooled water temperatures. These temperatures are shown in Table 2 for Tower No. 2.

Three condensing plants were taken into account and were designed to cope with each of the three cooling water quantities and have the necessary heat transfer surface to give final terminal differences ($T_3 - T_2$) of 6 deg, 10 deg and 14 deg F, T_3 being the vacuum steam temperature. The monthly averaged recooled temperatures, and the condenser overall terminal differences ($T_3 - T_1$) were established for each permutation of cooling tower size, water quantity and condenser design, and the averaged monthly vacuum for each arrangement established.

When the averaged monthly vacuum was established the differential variation in turbine performance was obtained from a vacuum correction curve similar to those those shown in Fig. 15.

The hydraulic losses of the cooling water systems were calculated, and the appropriate differential pumping costs included in the assessment, together with the differential values of turbine performance, and the differential capital cost of the various arrangements.

The cooling systems incorporating condensers designed for a terminal difference ($T_3 - T_2$) of 6 deg were found to be the most attractive.

The differential total costs of these systems are shown in Table 3 and in Fig. 16. A cooling tower duty D in the order of 60,000 was found to be the economic duty.

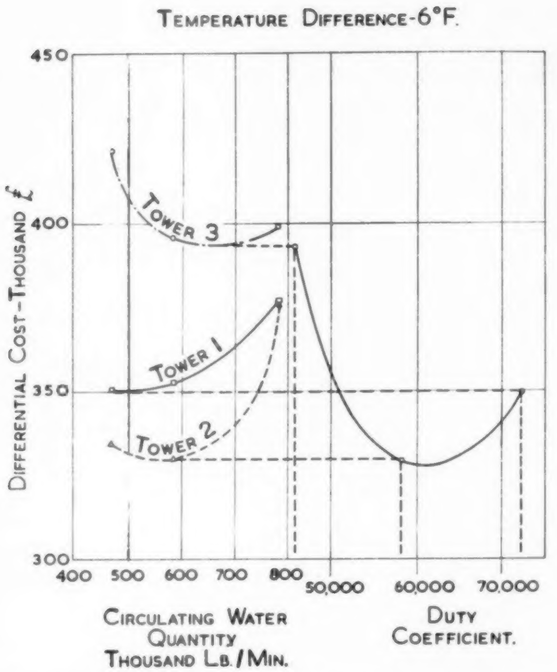


Fig. 16—Differential total costs of cooling systems with condensers having terminal differences of 6 deg F. (See also Table 3.)

This brief outline of the procedure for establishing the economic cooling tower size is based on an installation having no other cooling source than cooling towers, but in some locations it may be attractive to use a small river and supplement its cooling capacity with cooling towers. Under these circumstances the same approach is made to the problem, but the calculation become more complex in that the natural temperature, and water flow variations of the direct cooling source have to be taken into account.

In Britain the complete power station has generally been planned from the outset and the cooling water system has been fully interconnected so that any combination of cooling towers and pumps can operate with any condensing plant. This has been justified on the grounds that when a machine has been out of service the improvement in performance with the remaining machines, due to a better vacuum with all cooling towers in operation or reduced pumping power when relatively low recooled temperatures have been available, outweighs the cost of the interconnecting piping and valves. The effects of this feature, however, are not of a sufficient order for it to be necessary to take them into account in the initial economic study.

Fig. 17—Since the natural draft tower is a large body this view of several for a 300 Mw plant shows they can be arranged in an eye-pleasing form

Appearance

As the modern natural draft cooling tower is a relatively large structure it may be necessary to consider a group of towers from an aesthetic point of view and Fig. 17 is included to give some indication of appearance of a cooling installation suitable for 300-MW plant capacity.

Acknowledgments

In conclusion the authors would like to thank Mr. F. H. S. Brown, Member for Engineering of the CEGB for permission to publish this paper and also their colleagues who have contributed in any way to the preparation of this paper.

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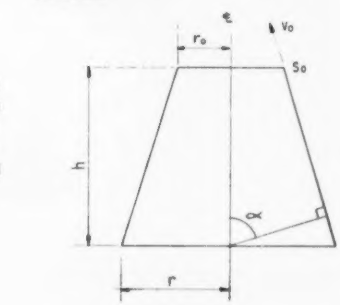
Appendix I—Analysis of "Diabolo" Shell

AN EXTENSION OF THE ANALYSIS APPLIED TO THE CYLINDER TO THE MORE COMPLICATED CASE OF THE TRUNCATED CONE WITH A DIRECT FORCE V_o AND SHEAR FORCE S_o ACTING ON THE TOP LEADS TO THE FOLLOWING RELATIONSHIPS:



$n = r_o/r$

$$V = -\frac{p''h^2(n+2n^2)}{6r_o \sin^3 \alpha} - \frac{p \cos \alpha\, h(1+n)}{2 \sin^2 \alpha} - \frac{s_o'(n-n^2)}{\cos \alpha} + V_o n$$

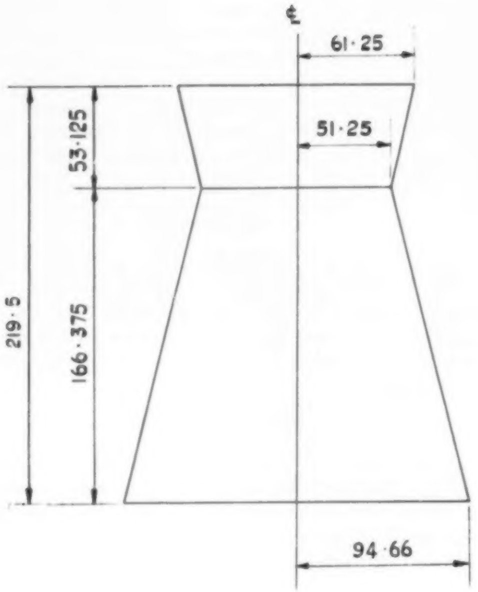
$$S = \frac{p'h(1+n+n^2)}{3 \sin^2 \alpha} + S_o n^2$$

$$t = -pr \operatorname{cosec} \alpha$$

A SUPERIMPOSED SECTION HAVING A DIFFERENT ANGLE OF INCLINATION α AND WITH DIRECT AND SHEAR FORCES AT ITS BASE V_1 AND S_1 WILL PRODUCE THE FORCES V_o AND S_o AS FOLLOWS :-

V_o = THE COMPONENT OF V_1 IN THE DIRECTION OF THE LOWER SHELL

S_o = S_1 - THE FIRST DIFFERENTIAL WITH RESPECT TO β OF THE COMPONENT OF V_1 IN THE HORIZONTAL DIRECTION.



TOP SECTION

$\alpha = 100°36'$, $\cos\alpha = -0·184$, $\sin\alpha = 0·983$,

$\sin^2\alpha = 0·966$, $\sin^3\alpha = 0·950$, $h = 53·125$,

$h^2 = 2822$, $n = 1·195$, $n^2 = 1·428$,

$(1+n) = 2·195$, $(1+n+n^2) = 3·623$,

$(n+2n^2) = 4·051$, $r_0 = 61·25$.

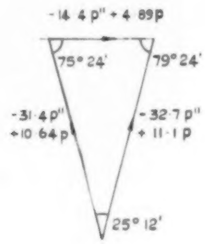
$$S = +\frac{p'h\,(1+n+n^2)}{3\sin^2\alpha}$$

$$= +p'\,\frac{53·125\times 3·623}{3\times 0·966} = \underline{\underline{+66·4\,p'}}$$

$$V = -\frac{p''h^2(n+2n^2)}{6r_0\sin^3\alpha} - \frac{p\cos\alpha\,h\,(1+n)}{2\sin^2\alpha}$$

$$= -p''\,\frac{2822\times 4·051}{6\times 61·25\times 0·950} + p\,\frac{0·184\times 53·125\times 2·195}{2\times 0·966}$$

$$= \underline{\underline{-32·7\,p'' + 11·1\,p}}$$



LOWER SECTION

$\alpha = 75°24'$, $\cos\alpha = 0·252$, $\sin\alpha = 0·968$,

$\sin^2\alpha = 0·937$, $\sin^3\alpha = 0·907$, $h = 166·375$,

$h^2 = 27580$, $n = 0·542$, $n^2 = 0·294$,

$(1+n) = 1·542$, $(1+n+n^2) = 1·836$, $(n+2n^2) = 1·13$,

$(n-n^2) = 0·248$, $r_0 = 51·25$.

$$V_0 = -31·4\,p'' + 10·64\,p$$

$$S_0 = +66·4\,p' - (-14·4\,p''' + 4·89\,p') = +14·4\,p''' + 61·5\,p'$$

$$S = \frac{p'h\,(1+n+n^2)}{3\sin^2\alpha} + S_0 n^2 = p'\,\frac{166·375\times 1·836}{3\times 0·937} + S_0 n^2$$

$$= 108·6\,p' + (18·1\,p' + 4·23\,p''') = \underline{\underline{126·7\,p' + 4·23\,p'''}}$$

$$V = -\frac{p''h^2(n+2n^2)}{6r_0\sin^3\alpha} - \frac{p\cos\alpha\,h\,(1+n)}{2\sin^2\alpha} - \frac{S_0'(n-n^2)}{\cos\alpha} + V_0 n$$

$$= -p''\,\frac{27580\times 1·13}{6\times 51·25\times 0·907} - p\,\frac{0·252\times 166·4\times 1·542}{2\times 0·937} - 0·985\,S_0' + 0·542\,V_0$$

$$= -111·6\,p'' - 34·6\,p - (60·7\,p'' + 14·2\,p^{iv}) + (-17·0\,p'' + 5·8\,p)$$

$$= \underline{\underline{-28·8\,p - 189·3\,p'' - 14·2\,p^{iv}}}$$

McCone Contrasts Russian Nuclear Planning

"I feel compelled," stated John E. McCone, AEC, upon his return from a tour of Russia, "to discuss one aspect of the Soviet effort of great interest to me. It has to do with the speed with which the Soviets appear able to reach a specific objective once they decide that it is of overriding importance to them... At the Soviet atomic laboratory at Obninsk, near Moscow, we were shown an advanced fast neutron experimental reactor fueled with plutonium oxide. There are no such reactors fueled with plutonium in this country nor elsewhere in the world. We were told that this plant was designed and constructed in a year's time.

"As a second example we were told that in ten months, Soviet scientists designed and built the giant controlled thermonuclear experimental machine known as the OGRA.

"From what I observed of Soviet dedication, their energy, and their competence, I have no reason to doubt these statements. In our country, under procedures that perhaps have become traditional, a like undertaking could take a far longer time."

Speaking of American ability to achieve results Mr. McCone continued, "This we did with great success in the last war. Let no one mistake that a decisive struggle of quite a different nature is going on right now. We must adopt some of the same philosophies of our war-time practices if we are to win the contest for technological and economic superiority in which we presently are engaged... Be prepared to seize upon the undertakings which are of importance and advance them on an urgent basis. If we do less, our position of leadership soon will fade away.

I do not want to be misunderstood on this point. Freedom in this country must not be subordinated. We must not lose sight of our ultimate purpose to preserve a form of government in which the dignity of the individual and his traditional freedoms are recognized."

Abstracts From the Technical Press—Abroad and Domestic

(Drawn from the Monthly Technical Bulletin, International Combustion, Ltd., London, W. C. 1)

Fuels: Sources, Properties, Preparation

Coal—Its Rank and Properties. J. P. Lauder. *Engng. Boil. Ho. Rev.* 1959, **74** (July) 210–2.

The N.C.B. classification of British coals, types of coal, coking properties and approximate output of each are tabulated. The coal sizes and ranks most suitable for burning on Low Ram, chain-grate and underfeed stokers are briefly indicated.

Evaluation of Low-Quality Coals as Boiler Fuel. H. Werkmeister. *Elektwirtsch.* 1959, **58** (June 20) 417–23 (in German).

On the basis of a questionnaire UNIPEDE has carried out an analysis of the evaluation of low-quality coals which has resulted in an equation of low-quality coals which has resulted in an equation for the evaluation of a coal in which the utilization value of a coal is a direct function of its calorific value; this is, however, modified by economic factors variable from country to country. This new evaluation function is compared with those valid in other countries.

The Tensile Strength of Coal. R. Berenbaum and I. Brodie. *J. Inst. Fuel* 1959, **32** (July) 320–7.

A method particularly convenient for measuring the tensile strength of coal is described and results obtained with it are presented and discussed.

Investigations of Bituminous Coals of the Chalk and Tertiary Formation. II. Plasticity Properties and Coking Characteristics of the Chalk-Tertiary Coals—Influence of Petrographic Composition of these Coals on Their Rheological Behavior. H. Hoffmann and K. Hoehne. *BrennstChemie* 1959, **40** (July) 219–27 (in German).

The coals mentioned in the first part were studied with regard to plasticity under carbonizing conditions. The relationship between petrographic composition and rheological behavior is presented in graphs and tables.

The Chemistry of Carbonization, Studied on Polymeric Bituminous Coal Model Substances. III. Synthesis of Model Substances. P. M. J. Wolfs, D. W. van Krevelen and H. I. Waterman. *BrennstChemie* 1959 **40** (July) 215–9 (in German).

The methods used for the production of various model substances are described and the behavior of these substances during heating to 200–250 C discussed.

Heat: Cycles and Transmission

Radiant Interchange within an Enclosure. J. T. Bevans and R. V. Dunkle. *A.S.M.E. Preprint No.* 59-HT-4 1959, (Aug.) 19 pp.

A mathematical study in three parts: 1. Absorption and emission behavior of gases; 2. General interchange equations; 3. A method for solving multinode networks and a comparison of the band energy and gray radiation approximations. It is shown that the band energy method is more accurate though more time consuming. It is stressed that many assumptions had to be made because of lack of reliable data.

Heat Transfer to Water Flowing Parallel to Tube Gundles. J. Weisman. *Nuci. Sci. Engag* 1959, **6** (July) 78–9.

It is shown that square and triangular pitch lattices when plotted as function of ratio of water flow area to the total cross-sectional area of an infinite lattice yield essentially the same heat transfer coefficients at equivalent ratios. This means that lower fluid velocities can be used to obtain high heat transfer coefficients.

Steam Generation and Power Production

Experimental Determination of Specific Volumes of Steam at High Temperatures and Pressures. V. A. Kirillin and S. A. Oulibin. *Teploenergetika* 1959, (Aug.) 71–3 (in Russian).

New data for the temperature range 500–620 C and up to 700 kg/cm² are presented.

Steam Plant Enclosures May Be Justified. C. D. Birget and J. H. Kline. *Electr. Wrld.* 1959, **152** (Aug. 3) 46–8.

The calculations show that in the northern regions of the United States a fully enclosed power plant will be cheaper in overall costs than a semi- or fully-outdoor plant while in the southern regions the cost differences would be small.

The Calculation of Static Heads and Hydraulic Resistances Occurring in Vertical Risers during Movement of a Steam-Water Mixture. V. A. Lokshin and A. L. Shvartz. *Teploenergetika* 1959, (Aug.) 73–7 (in Russian).

Experimental data on useful heads of steam-water mixtures in risers are presented in graphs.

Operating Characteristics of Boilers in Relation to Steam Demand. E. G. Ritchie. *J. Inst, Fuel* 1959, **32** (July) 314–6.

Installation of thermal storage systems (by pressure drop, steam accumulators, hot process water storage, feedwater accumulator) in smaller industrial plants with widely fluctuating steam demand permits a uniform firing rate of the boiler with consequent efficient combustion and absence of smoke and grit emission.

Forward Planning for the Industrial Plant. C. E. Miller. *Coal Utilization* 1959, **13** (July) 25–8.

Because of changing fuel costs and availability it is recommended to install boilers able to efficiently burn solid liquid and gaseous fuels. Examples of boilers specially designed for rapid conversion are illustrated.

Thermal Liquids Match Process Heating Needs. R. C. Bellas. *Power* 1959, **103** (Aug.) 53–8.

The reasons for an advantages of using liquid heat transfer media for heating purposes in various industrial processes are set out, properties and operating limits of some of these media presented in tables and graphs and boiler plant specially designed for the purpose illustrated. One of the most important points is the prevention of overheating; the means adopted in various design are described.

Some Design Features of the Meramec Twin Furnace Reheat Steam Generator. M. F. Dallen and D. B. Stewart. *A.S.M.E. preprint* No. 59-SA-68 1959, (June) 13 pp.

This boiler is rated at 1850 klb/h at 2150 psi and 1010/1010 F, of semi-outdoor construction, twin furnaces with one convection pass containing the reheater, the other the final superheater. Reheat temperature control is by bias-firing of the furnaces, superheat temperature control by condenser type heat exchange with feedwater. Three mills each of 9700 lb/h capacity are connected to two exhausters each to provide for the bias firing. The burners are surrounded by gas rings for firing natural gas alone or in any ratio with coal. The dual circulation principle is applied to reduce the silica content of the steam.

Solid Fuel Firing

Combustion of Crushed, Dried Texas Lignite and Char in Steam-Power Boilers. V. Z. Caracriti and H. D. Mumper. COMBUSTION 1959, **31** (July) 34–44.

A detailed account is given of the design of the slagging furnace boilers (800 klb/h, 1550 psi, 1005/1005 F) and the modifications found necessary for the continuous firing of this fuel. Also discussed are furnace performance, ash deposition characteristics, chemical composition of the ash, fuel abrasion, firing of lignite char and the effects of flue gas as a primary fuel-transport medium.

Liquid and Gaseous Fuel Firing

Comparison of Different Methods of Fluid-atomising Oil Flames and the Effect on Flame Emissivity and Radiation of the Addition of Carbon Black to Liquid Fuels. Performance Trial No. VIII of the Joint Committee of the International Flame Research Foundation on the Experimental Oil and Gas Furnace at Ijmuiden. 1. Comparison of Different Methods of Fluid Atomizing Oil Flames. 2. The Effect on Flame Emissivity and Radiation of the Addition of Carbon Black to Liquid Fuels. E. H. Hubbard. *J. Inst. Fuel* 1959, **32** (July) 328–43.

In the first part differences in the use of a small nozzle at high pressure, a convergent-divergent nozzle or the addition of preheat to the atomizing fluid were investigated; all have little effect on the flame. The total radiation of flame atomized by air or steam is the same but the former is more evenly distributed over the length and its temperature is higher. Addition of a 50/50 suspension of carbon black to gas oil and fuel oil increased flame emissivity roughly in proportion to the increase in C : H ratio. The difficulties in keeping more than 6 per cent C by weight in suspension are discussed.

Water-Side Corrosion and Water Treatment

The Removal of Organic Substances from Raw Water by Means of Special Filter Aids. H. Hanning. *Mitt*

V.G.B. No. 60 1959, (June) 179–81 (in German).

Active carbon has proved in numerous experiments to be the best means of removing organic substances, even phenols and phenolic chlorides being completely removed. The economics of this method are discussed.

Complete Demineralisation and Organic Substances. C. Janssen. *Mitt. V.G.B.* No. 60 1959, (June 172–8 (in German).

The reduction in capacity of strongly basic ion exchanger caused by the presence of organic substances in the raw water is discussed. Strongly acid ion exchangers on styrol basis are little affected but those on a carbon basis to a large extent. Treatment of the ion exchanger with sodium chloride and its addition to the regenerant is recommended as one means of restoring the initial capacity. Others are the pretreatment of the water with lime, by flocculation or in an adsorption filter of special porous resins.

Desilicizer Feedwater Process Aids Scovill. C. K. Stickney, W. C. Burns and J. W. Walter. *Pwr. Engng.* 1959, **63** (Aug.) 93–4.

It has been found that strongly basic anion exchanges operating on the hydroxyl cycle are capable of removing silica from soft water and a cheaper process than hot-process softening or demineralization. At the Scovill plant the residual silica content is less than 1 ppm and has eliminated HCO_3 and CO_3 alkalinity. Further advantages are that shells, valves and pipelines do not need lining, that maintenance is low, and that it can be added directly to existing sodium-zeolite softeners.

Filming Amines Control Corrosion in Utility Plant Condensate System. E. E. Galloway. *Corrosion* 1959, **15** (Aug.) 99–100, 103.

Tables are presented showing the corrosion rate of iron in the condensate lines before and after introducing octadecylamine at the rate of 5 ppm into the feedwater. The protective film formed prevented attack by oxygen and carbon dioxide in the condensate lines but did not protect wet areas of the turbines. The filming amine has no adverse effect on copper or copper alloys.

Flue Gas, Ash and Dust

The Application of a Mechanical Collector in Combination with an Electrostatic Precipitator. A. B. Walker and J. Phyl. *Blast. Furn. Steel Plt.* 1959, **47** (June) 622–4.

The factors are discussed on which

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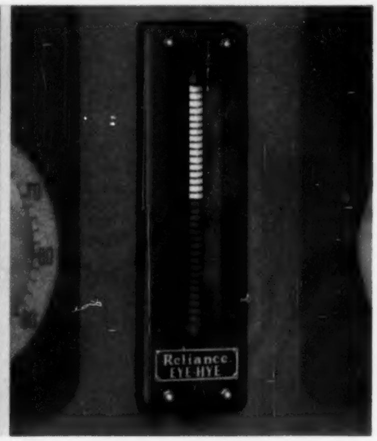

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the decision must depend whether the mechanical collector is to be installed upstream or downstream of the precipitator.

Possibilities of Fly Ash Separation in Mechanical Separators Taking into Account Experience with Slagging Furnaces. R. Nagel and R. Ibing. *Mitt. V.G.B.* No. 60 1959, (June) 220–9 (in German).

The separation efficiency of cyclone separators and its relationship to costs, power consumption and pressure drop are discussed. High duty cyclones have been developed able to reduce the fly-ash content to below the limit set by law. Results of tests are presented. Deposit formation and complete clogging have been experienced, especially in boilers with slagging furnaces when the temperature of the flue gas fell below the dew point. Where cyclone separators and electrostatic precipitators are used it is suggested that better results are obtained if the former are installed downstream of the precipitators.

Recent Developments of Electrostatic Precipitator Plants. H. Brandt. *Mitt. V.G.B.* No. 60, 1959, (June) 229–35 (in German).

The variables on which the separation efficiency of electrostatic precipitators depends are the electric charge the fly ash and gas obtain in the furnace (this may be positive or negative), the gas velocity and composition (moisture content, SO_3 content, SO_3/SO_2 ratio), shape of ash particles, their dielectric constant and composition, design of electrodes and distance between electrodes. In plants burning bituminous coals good experience has been obtained with sparking point electrodes made of flat, soft steel and precipitating electrodes in the shape of perforated boxes; rapping is preferred to vibrating; gas passages should have a width of 10 in.; gas velocity not above 2 m/s, preferably 1.5 to 1.8 m/s; high voltage supply either a.c. or equipped with transducers.

Heat Recovery Plant

Auxiliary Steam System to Blow Soot. Anon. *Electr. Wrld.* 1959, **152** (Aug. 17) 76.

At the Eddystone supercritical pressure plant three oil-fired boilers each rated at 39,500 lb/h at 500 psi have been installed to supply the sootblowers.

Power Generation and Power Plant

Efficiency of Steam Generation. J. N. Williams. *Pwr. and Wrks. Engng.* 1959, **54** (July) 395–401.

The "indirect" method of determining the efficiency, the Siegert formula for dry heat loss, gas temperature and CO_2 determinations are described.

The Economics of Air Preheat on Industrial Boilers. C. L. Brown. Combustion 1959, **31** (July) 51–5.

The improvements in boiler efficiency by the installation of air preheaters is assessed on the basis of two typical industrial plants with installation costs set against fuel savings. Formulæ are presented for making the efficiency calculations.

Ventilation of a Steam Power Station. A. G. Kelly. *Engineering* 1959, **188** (Aug. 14) 42–4.

The problem of economic and efficient ventilation of a steam power station is set out and tests carried out on a station with 60 MW of plant in service described. The results are discussed.

Thermionic Converters. K. G. Hernquist. *Nucleonics* 1959, **17** (July) 49–53.

Advances in the design of thermionic cells are described which promise very good efficiencies in the direct conversion of the thermal energy of nuclear reactors into electricity at high temperatures.

The Fuel Cell. H. A. Liebhafsky and D. L. Douglas. *Mech. Engng.* 1959, **81** (Aug.) 64–8.

A review of the different approaches to the design of an efficient fuel cell and of recent progress by leading specialists.

Fluid Coke is Primary Fuel in New Avon Steam Plant. H. B. De Benedetti, F. H. Farmer and E. F. Courtney. *Pwr. Engng.* 1959, **63** (Aug.) 68–71.

The fluid coke at a temperature of 200–250 F is directly delivered by air to the boilerhouse bunkers and then pulverized in attrition-type mills lined with tungsten carbide. The rotating elements are also faced with this material. The pulverized coke is passed to the opposed burners tilted townward at an angle of 35° in the boiler rated at 500 klb/h at 600 psi and 750 F or 1300 psi and 950 F. Part of the ash is collected as slag in the furnace bottom; fly ash from the cyclone separators is mixed with the coke and passed again through the mills. The abrading action of unburnt coke particles assists in keeping down deposit formation. Up to 90 per cent of coke in a mixture of coke and gas or oil has been fired successfully.

Meramec 3 Steam Electric-Generating Unit. C. H. Rulfs and J. K.

Bryan. *A.S.M.E. Preprint* No. 59–SA-27 1959, (June) 8 pp.

The unit cost $40 million and the fuel cost over the life of the unit is estimated at 22¢ per million Btu; the load factor should average 75 per cent. The area required by the 125 MW units 1 and 2 was 215 sq ft/1000 kW, for the 250 MW unit 3 was 164 sq ft/1000 kW and for the 300 MW unit 4 it will be 134 sq ft/1000 kW. The turbine is divided into a hp and ip unit running at 3600 rpm and a lp unit running at 1800 rpm which has an axial flow design resulting in a higher temperature of the condensate returned to the feedwater cycle. Three half-size boiler pumps have been installed. The dual circulation principle of the boiler is expected to reduce boiler blowdown by 80 per cent.

The "Schwarze Pumpe" Power Station. 1st Stage. K. Bruche. *Energietechnik* 1959, **9** (Aug.) 341–52 (in German).

In the East-German Republic a large combined undertaking is being developed on brown coal basis, including briquetting works, coke ovens, gas works and power stations. The power stations will have to provide energy for three briquetting factories with an output each of 2.1×10^6 t/a, one pressure gasification plant for 3.5×10^9 m^3/a of gas, three coking plants each for 0.9×10^6 t/a high temperature coke and 0.5×10^9 m^3/a coke oven gas, and also steam for process and space heating and a district heating network. The first stage of its power station "West" contains 230 t/h boilers, those of the 2nd and 3rd stages will contain units consisting of a 420 t/h boiler and a 50 MW extraction-back-pressure turbogenerator. There are 6 boilers each rated at 230 t/h at 125 atu and 535 C and a feedwater entry temperature of 235 C, three 50 MW extraction-back-pressure turbogenerators and four 25 MW condensing turbogenerators. Hot water for process heating is supplied at 110/180 C from calorifiers by exchange with steam from the 17.5 ata mains, hot water at 70/130 C for space heating by exchange with steam at 5 ata. The boilers are of the semi-outdoor type. The brown coal contains 57.5 per cent moisture, 7.5 per cent ash and 1 per cent S, pulverized in mills each of 32 t/h capacity and injected through corner burners. Bottom ash is removed hydraulically, fly ash from the hoppers of the electrostatic precipitator, flue gas ducts and chimney bottom pneumatically. The description also includes boiler controls, turbogenerators, feedwater treatment and 30 kV switching station.

Sound-proofing Measures in the Building of the Heat-Power Stations. L. Cremer and A. Müller. *Energie* 1959, **11** (July) 311–3 (in German).

Since the two Munich heat-power stations are sited in the center of the town particular emphasis was laid on sound proofing the buildings and machinery. The problems encountered and solutions employed are given.

Insulation of a Nuclear Power Station. Anon. *Nucl. Energy Engr.* 1959, **13** (Sept.) 456–7.

The difficulties of insulating the pressure vesels at Berkeley nuclear power station owing to the large expansion are discussed and the methods applied are described.

N.S. Savannah. Reactor Physics and Core Design. P. M. Wood and Z. Levine. *Nucl. Energy Engr.* 1959, **13** (Sept.) 450–5.

The report describes the fuel elements and their arrangement in the core, and the reactivity calculations.

Operating Experience at Vallecitos. Anon. *Pwr. Engng.* 1959, **63** (Aug.) 48–9.

This boiling water reactor supplies steam directly to a 5 MW turbogenerator but is mainly used for testing purposes, such as the fuel elements for the Dresden nuclear reactor. Radiation levels in the turbine and reactor building are low, the maximum exposure for any individual did not exceed 390 mr in one and a half year. Turbine and generator are thus accessible at all times, but activity levels at the demineralizer, air ejector and steam purifier mainly due to N 16 are higher. Stability of the reactor has been very good at pressures of 900–1000 psi and outputs up to 30 MW (thermal) and at load increases from 0 to full load in less than 1/2 minute.

Materials and Manufacturing Processes

Steel for Reactors and Processing Plants. B. Watkins. *Nucl. Engng.* 1959, **4** (July/Aug./Sept.) 296–303.

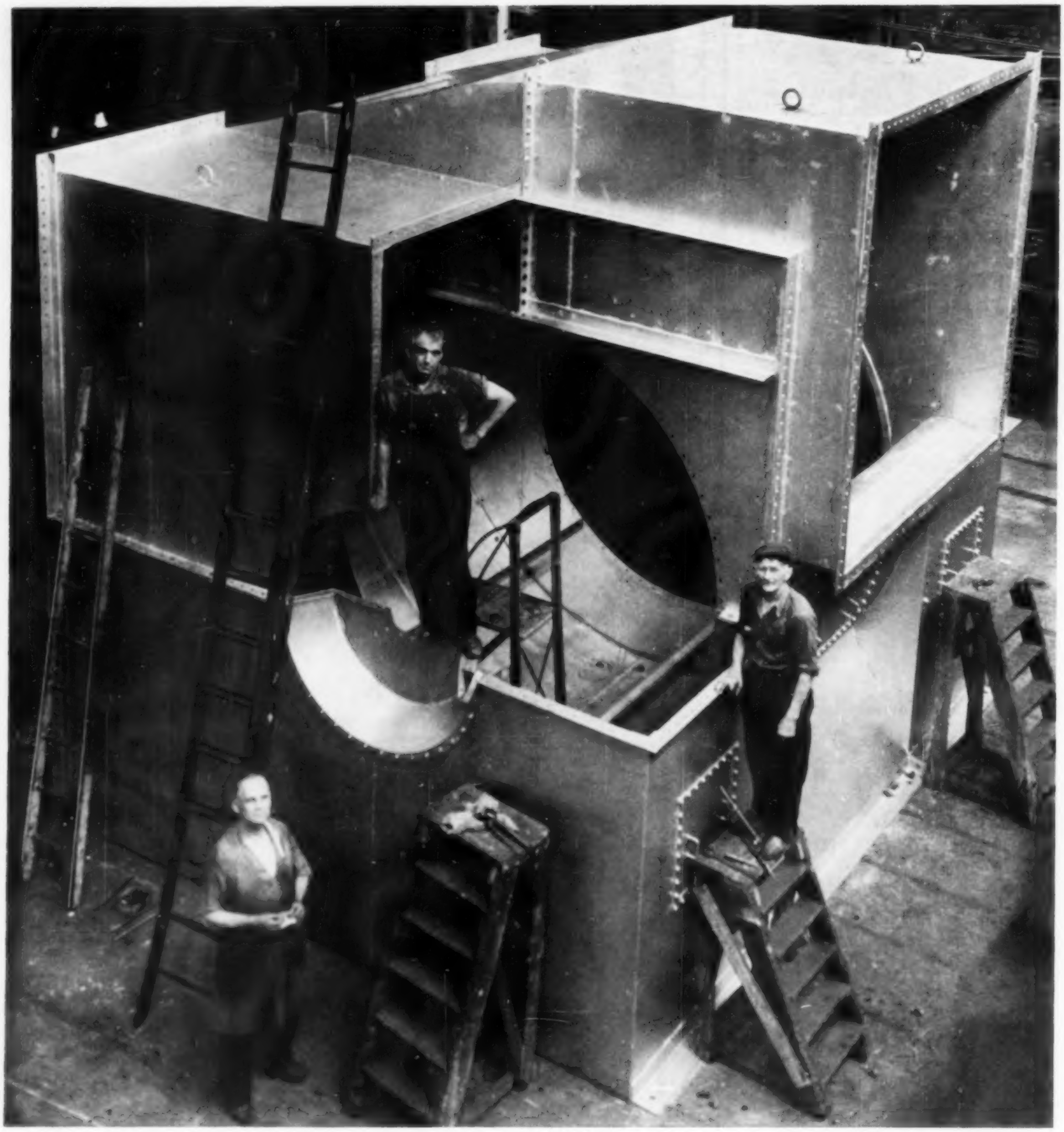
A review of the special properties essential for application of mild steel in nuclear reactors under the headings: 1. Physical properties; 2. Mechanical properties (tensile, creep); 3. Chemical compatability; 4. Welding; 5. Nuclear considerations. A data sheet is included.

New Ceramic Coatings. Anon. *Nucl. Energy Engr.* 1959, **13** (Sept.) 448–9, 454.

The composition of the new coatings for steels in nuclear reactors developed by the U. S. Bureau of Standards is tabulated and the properties are discussed.

Advertisers' Index

(Continued on page 63)



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